POWER

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Pigs in Clover

Power and Refrigeration for Model Ice Cream and Dairy Plant

Modern 450-ton Three-Unit Refrigerating Plant Using High-Speed Two-Cylinder Ammonia Compressors, Cross-Connected So That Either Cylinder May Be Used on High or Low-Temperature Service—Power and Lighting Are Supplied by a 450-Kw. Electric Generating Condensing Plant, the Cooling Water Being Conserved by a Large Cooling Tower on the Roof of the Building

ONE of the largest ice-cream and dairy plants in the country is being operated in Cincinnati, Ohio, by the French Bros. Bauer Company. It has now been in operation for about a year, being a consolidation of two former plants, one engaged mainly in the production of ice cream and the other in the manufacture of butter and pasteurization and distribution of milk and

FIG. 1. ICE-CREAM HARDENING ROOMS, CAPACITY, THIRTY-THREE THOUSAND GALLONS

cream for Cincinnati and the surrounding territory. Both of the plants also manufacture ice, but not in sufficient quantity for packing the ice cream during the summer months.

As each of the old plants required a full force of operators, this naturally made the cost of production higher than would have been the case in one plant, and as the equipments had become inadequate, it was decided to build a new plant of sufficient capacity for the combined output of the two old ones, with a reasonable allowance for future increase.

The building, constructed of reinforced concrete throughout, was erected on the site of an old brewery. The underground cellars, with their heavy arched ceilings, retaining walls and many intricate underground passages, offered an interesting problem in the design of the foundations and basements, with the purpose of reducing the excavation work and new foundation walls to the minimum. All reinforced-concrete floors throughout the building, where considerable water is used on the floor, are waterproofed with three-ply membrane waterproofing and, with some exceptions, finished on top with three inches of concrete. In the bottle-filling and ice-cream making departments the floors, after having been waterproofed, are finished with Terrazzo, and for the shipping platform the finish is vitrified brick paving. All cold-storage rooms, as well as the ice tanks, are insulated with corkboard with all exposed surfaces on walls and ceilings plastered with portland-cement plaster. The exterior of the building is finished with red brick facing between concrete beams and columns at each floor, and the building is trimmed with terra cotta.

To give some idea of the power and refrigerating requirements and the general arrangement, a brief survey of the various departments may be of interest. At the two basement levels, the lower being 28 ft. below the sidewalk, are the boiler and engine rooms, ice tanks, daily ice storage and a 3,000-ton season ice-storage room. On the first floor of the building is the bottled-milk storage room, ice-cream-can washing room, ice-cream-making and the ice-cream hardening department and the loading platform. The driveway is under the building proper along the east and south sides. Being about 30 ft. wide, the long span over the driveway necessitated on each floor above it beam and girder

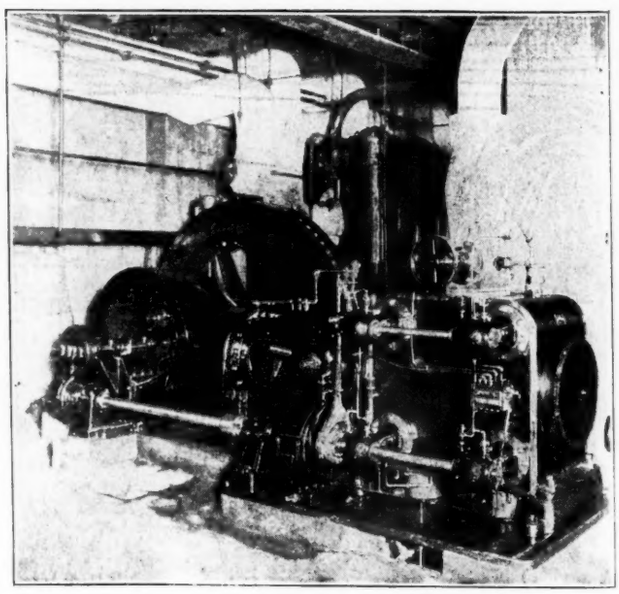

FIG. 2. ONE OF THE 150-TON REFRIGERATING UNITS. VERTICAL HIGH-SPEED COMPRESSOR IN BACKGROUND

construction, while the rest of the building is flat-slab construction with the columns placed on 25-ft. centers.

Under the south driveway is the salt bin, from which salt is elevated to hoppers on the second floor and spouted to the first floor as required for the packing of ice cream in tubs.

One-half of the second floor is taken up by general offices and the remainder of the space by the bottle- and can-washing departments, the bottle-filling depart-

ment, the ice-cream mixing department and the cold-storage rooms for fruits and other supplies used by the ice-cream mixing departments. On the third floor one-half of the space is utilized for storage purposes and the other half for the butter-making department, butter cold-storage rooms and for pasteurizing milk and cream. The fourth floor also has storage space and, in addition, the milk-receiving department and chemical laboratory.

From the foregoing it will be noticed that the various departments are so located in the building that after the milk and cream are received on the fourth floor, they follow through the manufacturing process by gravity until the products are ready for distribution on the first floor.

Milk and cream are received at the plant in cans, which are raised to the fourth floor by a continuous can elevator provided with an automatic loading table on the first floor and an automatic discharge table on the fourth floor. An electric high-speed platform elevator serves as a reserve unit in case of breakdown.

On the fourth floor the milk cans travel by gravity on a roller conveyor to the milk-weighing tanks. After the milk is emptied, the empty cans travel to the second floor to the can-washing department on a spiral conveyor, and after being washed they are discharged to the first floor on another spiral conveyor.

The milk and cream from the scale tanks are pumped to a holding tank and, after having been clarified, go to the milk and cream storage tanks, then to the pasteurizers and to cooling and holding tanks on the third floor. The milk and cream are bottled on the second floor, loaded into cases and sent to the milk-storage room on the first floor on a spiral conveyor.

After having been pasteurized, the cream for ice-cream making flows by gravity to the ice-cream mixing tanks on the second floor and again gravitates to the ice-cream freezers placed on the first floor immediately below the mixing tanks. The sour cream for butter making is elevated to the fourth floor, the same as the sweet milk and cream, is weighed in a separate scale tank and then pasteurized, ripened and made into butter on the third floor.

The mechanical equipment and refrigerating plant were designed for a maximum output in eight hours per day of pasteurizing and cooling 13,000 gal. of milk and sweet cream, 10,000 gal. of sour cream, the making of 25,000 lb. of butter, the freezing and hardening of 7,000 gal. of ice cream, together with maintaining necessary temperatures in cold-storage rooms and the making of 100 tons of ice per day.

A careful analysis of the load factor during the various months of the year in the older plants and as estimated for the different periods of the day based on the new output disclosed the fact that the maximum load would obtain for about seven hours of the day; that the load for the other seventeen hours would be about one-third of the maximum load, and that the average maximum load for the year would be about two-thirds of the maximum daily output. These interesting estimates made it appear advisable in so far as was possible to arrange the different mechanical units (boilers, generating units and ammonia compressors) into three of each kind, thereby giving flexibility in operation and making the best provisions for spare units and spare parts.

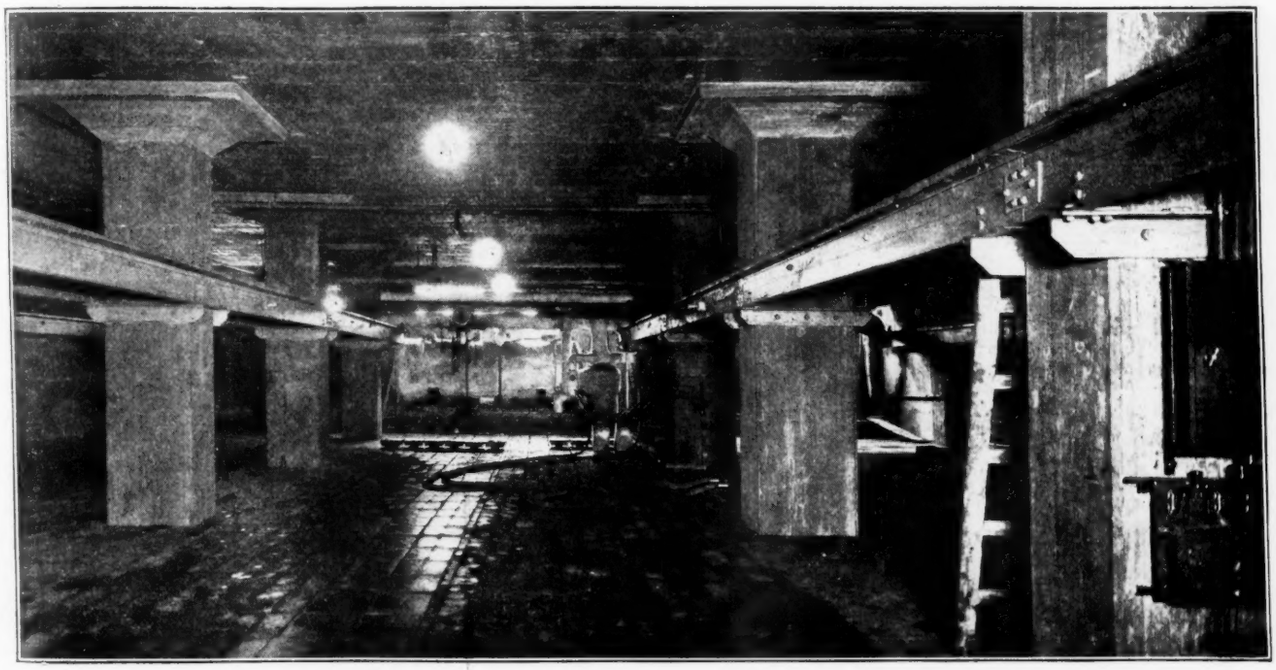

FIG. 3. VIEW OF THREE ICE-FREEZING TANKS EACH HAVING NOMINAL CAPACITY OF THIRTY TONS

Throughout the design of the plant the fundamental idea of arranging the process of manufacture to reduce the labor and power cost to the minimum was carried out, in that all products are handled by gravity in so far as is possible, and in selection of the mechanical equipment choice was made of the most efficient types for the purpose intended.

To conform with latest practice, high-speed ammonia compressors were installed. Efficiency was the primary factor in the selection, and space limitations compli-

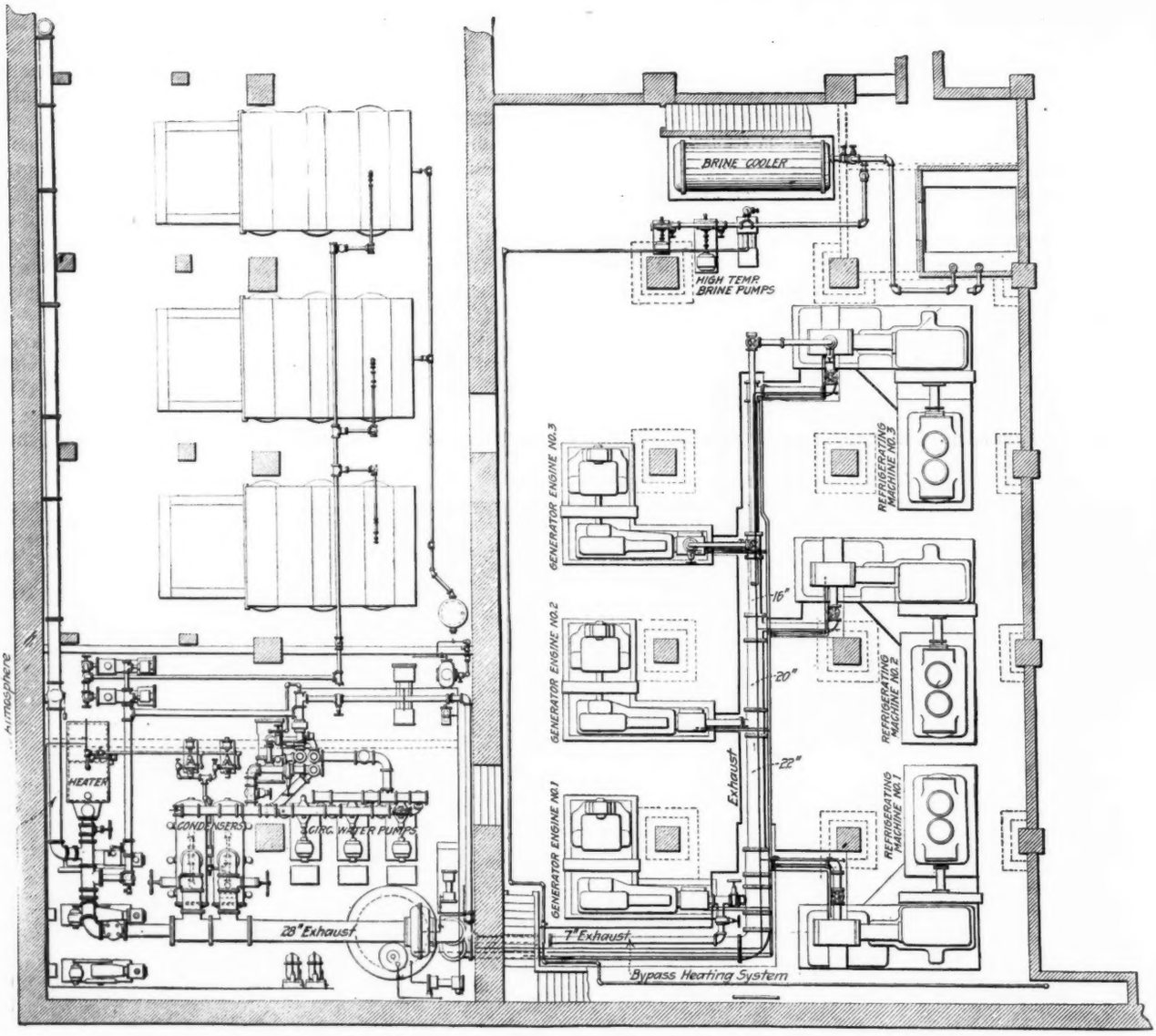

FIG. 4. GENERAL PLAN OF STEAM PIPING IN ENGINE, COMPRESSOR AND BOILER ROOM

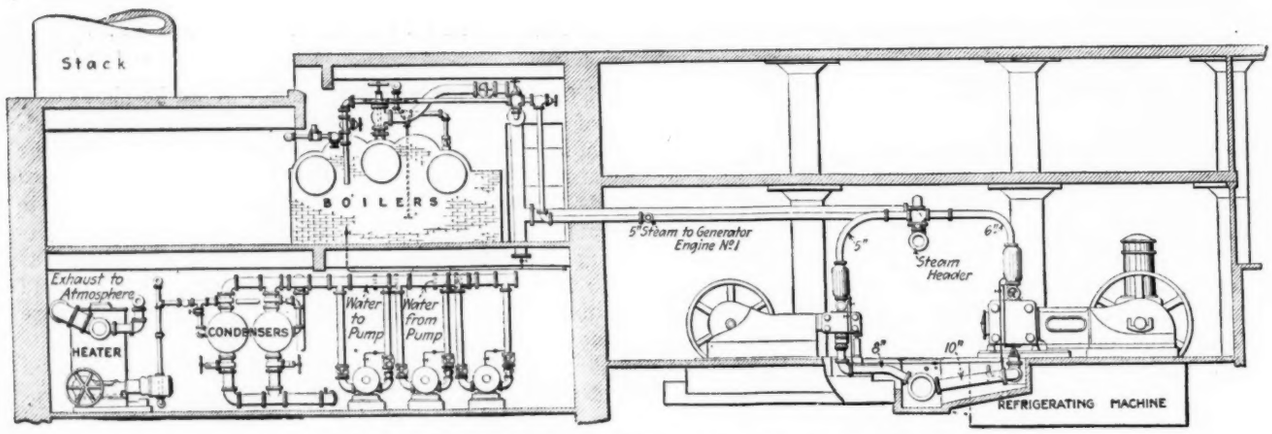

FIG. 5. SECTIONAL ELEVATION THROUGH POWER PLANT

cated by large building columns, necessitated vertical units. As intimated previously, there are three machines, each of the vertical, inclosed, single-acting, high-speed type, having two 18 x 20-in. cylinders and a speed of 150 r.p.m. The rating is 150 tons of refrigeration. A feature giving unusual flexibility to the operation is the connection of each cylinder to both high- and low-temperature lines so that a cylinder may be de-

voted to either service or both to the same service, as required. At times of light load one machine will take care of both classes of work, and when the load is heavy enough to require two machines, but perhaps not evenly divided, three of the four cylinders may be used in one service and the fourth cylinder in the other.

Horizontal poppet-valve, simple condensing engines, 20 x 24-in., drive the compressors. As will be evident from Fig. 2 a layshaft geared to the mainshaft operates

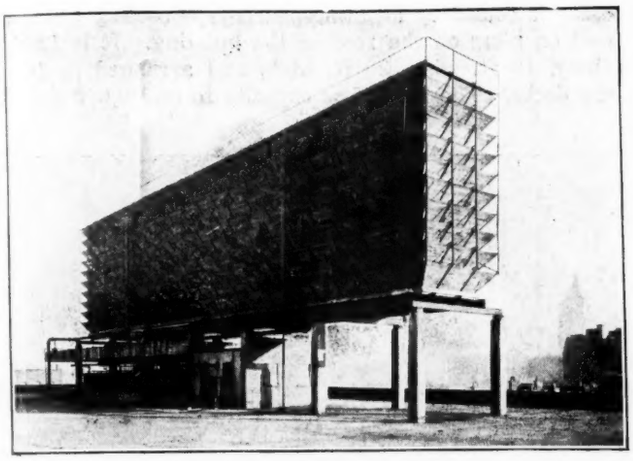

FIG. 6. AMMONIA FLOODED CONDENSERS AND COOLING TOWERS ON ROOF

the poppet valves, which are placed horizontally. Cams on this layshaft open the exhaust valves, and they are seated by springs provided in the bonnet. An eccentric and a secondary shaft open the steam valves at the top of the cylinder, the closing being effected by springs as in the case of the exhaust valves. For adjustment of load, the valve stems may be screwed in or out. A centrifugal governor on the layshaft controls the opening of the steam valves and consequently the speed of the unit.

Force-feed lubrication is provided for the steam cylinder and gravity oiling for the bearings. A centrifugal pump, belt-driven from the mainshaft, draws the oil from the crankpit and elevates it to a pedestal filter from which it flows back to the bearings. For the compressor lubrication there is also a belt-driven pump which pulls the oil from the crankcase and discharges it through a twin pressure filter to the seven bearings within the crankcase. There is thus continuous circulation of the oil, and electric alarms are provided to indicate any stoppage in the flow of oil.

Serving the ammonia compressors there are three banks of atmospheric condensers of the flooded type, aggregating 40 stands of 2-in. pipe, 12 high and 20 ft. long. The condensers, equipped with all necessary discharge, liquid, drain and pump-out headers, are located on the roof of the building.

For making ice there are three ice-freezing tanks, each 79 ft. long x 15 ft. 2 in. wide, arranged for 432 three-hundred-pound cans and piped with 10,000 ft. of 1¼-in. extra-heavy steel ammonia piping, the coils being arranged for the flooded system. Each tank is equipped with two 20-in. horizontal agitators, one agitator being placed at each end of the tank. Each agitator is belt-driven at 150 r.p.m. by a 3-hp. motor having a speed of 1,010 r.p.m. As all the ice is used for packing ice cream in tubs, it is made from city water and is opaque, no attempt being made to manufacture clear ice. The normal capacity of each tank is 30 tons.

To pull the ice, each tank is equipped with a one-ton electric hoist. The can is first dipped and the cake of ice delivered upon the dump, from which it slides through an automatic door into the daily storage of 500 tons in the first basement. From here the ice is conveyed either to the season storage of 3,000 tons' capacity or for immediate use to the first floor or to the ice crushers on the second floor. The ice is delivered to the crushers automatically, and the elevator cage returns to the basement automatically after having delivered the ice. The motors for the crushers and elevators are controlled from the shipping room on the first floor and are electrically interlocked, so it is impossible to operate the ice elevators without having first started the crusher. The ice from the crushers is spouted to the wagons in the driveway or to the shipping platform as desired.

All refrigerating work for cooling, holding vats, pasteurized milk and cream and for all refrigerated rooms for milk, cream and butter is done by means of brine circulation, for which is provided a shell-type brine cooler having 2,400 sq.ft. of exchange surface. The brine is circulated by means of two centrifugal brine pumps, brass-fitted throughout and direct connected to motors by means of flexible couplings. One pump has a capacity of 600 gal. of brine per minute and the other a capacity of 200 gal. per minute.

The fifteen 40-quart ice-cream freezers, chain-driven by 2-hp. motors, are provided with brine circulation, the brine being at a temperature of —10 deg. F. For this purpose is provided a shell-type brine cooler hav-

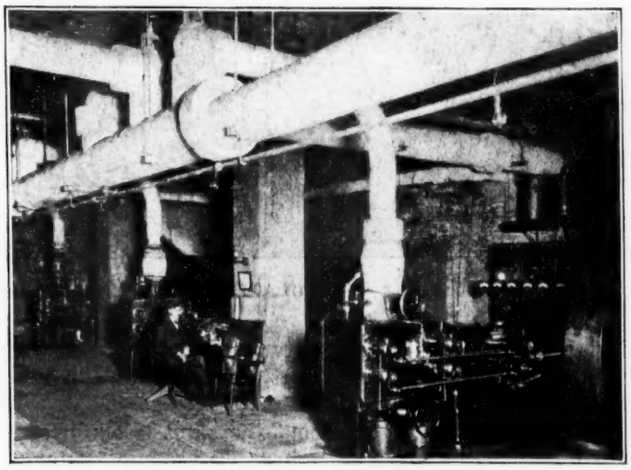

FIG. 7. JOHN O. BENNETT, CHIEF ENGINEER, IN HIS ENGINE ROOM

ing 750 sq.ft. of exchange surface, and the brine is circulated by means of a 300-g.p.m. centrifugal brass-fitted pump directly connected to a 1,700-r.p.m. motor.

After being frozen, the ice cream is hardened in a battery of nine hardening rooms arranged side by side and each connecting with a receiving vestibule on the north end and a delivery vestibule on the south end. These hardening rooms have a capacity for storing a maximum of about 33,000 gal. of ice cream at one time. Each hardening room is provided with an overhead

coil bunker arranged with baffles and direct-expansion coils over which the air circulation is blown by means of a 2,100-cu.ft. fan driven by a 2-hp. motor. The coils are arranged as a full gravity flooded system. From an accumulator 24 in. diameter by 8 ft. high, located on the second floor, the liquid gravitates to the coils. Provision is made for defrosting the coils of the hardening rooms either by passing the discharge ammonia gas from the refrigerating machines through the coils or by means of a brine spray over the coils. These two systems of defrosting can be used together or independently as desired.

The milk-storage room is refrigerated by air circulation from a coil bunker placed on a concrete platform in the corner of the room. The coil bunker is piped with 2,400 ft. of 2-in. galvanized brine piping, and the air is circulated by means of two fans of 3,000-cu.ft. per minute capacity, each direct connected to a 3-hp. motor.

All the dairy machinery about the plant is equipped with direct motor drive as well as all vacuum, water-circulating and brine pumps. An auxiliary steam driven pump for each service is, however, provided in case of emergency. The electric current for motors and lighting is generated by three 150-kw. 125/250-volt 200 r.p.m. three-wire direct-current generators connected to 15 x 22-in. simple condensing nonreleasing Corliss-type engines. Energy for power is taken off the outside wires at 250 volts; for lighting there is three-wire 125-volt distribution. A 2,000-ampere two-wire watt-hour meter measures the total energy consumed by the power circuits, and a 300-ampere three-wire watt-hour meter registers the current used for electric lighting.

As the engines driving the electric generators and ammonia compressors are operated condensing, two steam surface condensers have been provided, each having capacity to condense steam at the rate of 14,000 lb. per hour and to maintain a vacuum of 25 in. when supplied with 1,000 gal. of cooling water per minute at a temperature of 95 deg. As auxiliaries there are two motor-driven and one steam-driven dry-vacuum pumps and two motor-driven condensate pumps. The water of condensation is discharged into the feed-water heater.

Circulating water for the steam and ammonia condensers is cooled by an atmospheric-type cooling tower placed on piers on the roof of the building. It is 124 ft. long, 19 ft. wide, 30 ft. high, and arranged in 10 spray decks. The tower has capacity to cool 2,000 gal. of water per minute from a temperature of 130 deg. to within an average of about 4 deg. above the wet-bulb temperature, depending upon weather and wind conditions. The water from the cooling-tower pan passes over the ammonia condensers, from the ammonia-condenser pan drains to the steam condensers and is returned to the top of the cooling tower. The cooling water is circulated by means of three 5-in. centrifugal pumps each driven by a 40-hp. motor and having a capacity for handling 700 gal. of water per minute. Except for the distance from the roof to the top of the tower and the friction on the piping, the circulating system is balanced. An altitude valve controls the level of the water in the ammonia-condenser pan, the small amount of makeup water needed to overcome windage and evaporation being automatically admitted from the city mains.

The building is heated by exhaust steam during the

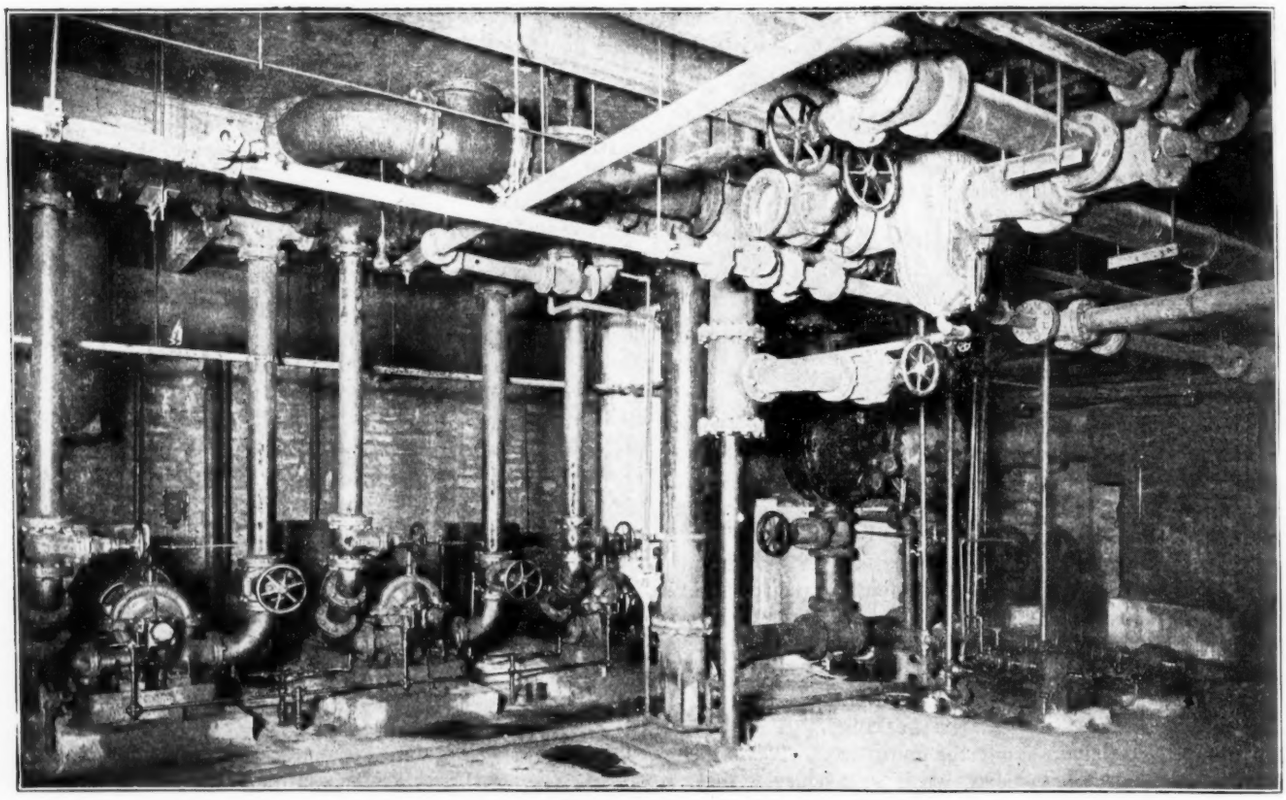

FIG. 8. CIRCULATING PUMPS, STEAM CONDENSERS AND FEED-WATER HEATER

winter months. The necessary supply for this purpose and for heating the feed water is obtained from the boiler-feed pumps, the auxiliary steam-driven brine and water pumps, provided as reserve units, the vacuum pump for the heating system and from one of the engines driving the electric generators, which is arranged with a bypass exhaust connection so that it may be operated noncondensing during the winter months.

In the boiler room as well as in other departments of the plant, the three-unit plan was followed, the equipment consisting of three Stirling boilers rated at 394 hp. and served by top-feed V-shaped stokers having approximately 80 sq.ft. of grate area, giving the usual ratio to steam-making surface of 1 to 50. The operating steam pressure is 145 lb. gage with no superheat. Through an 8-in. line provided with a main stop valve and an automatic nonreturn valve, each boiler is connected to a 12-in. main steam header, which is cross-connected to a distributing header of 10 and 8-in. diameter running lengthwise of the engine room. From either side supply lines lead to the generator and compressor units. The header is suspended from the ceiling and is anchored to prevent vibration. The engines exhaust into a common exhaust main increasing in diameter from 12 to 28 in. on its way through a floor trench to the condensers. From this line there is an atmospheric relief with connection to the feed-water heater, and as previously mentioned an independent exhaust or bypass from one of the engines to supplement the exhaust-steam supply for heating purposes.

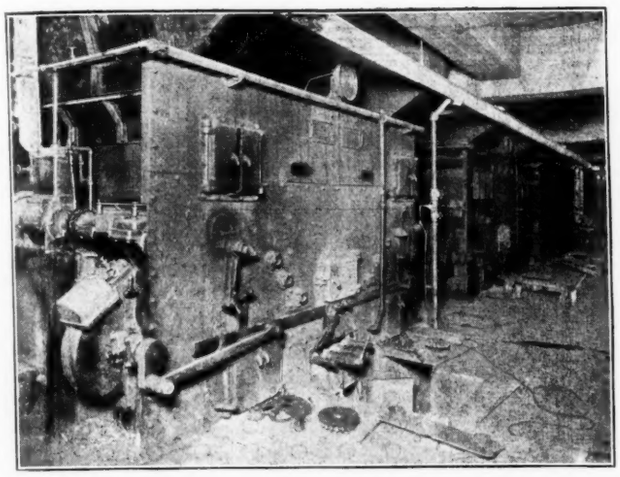

FIG. 9. VIEW OF TOP-FEED STOKERS AND COAL FEEDS FROM OVERHEAD BUNKER

A tapered breeching receiving the gases from each boiler leads to a reinforced-concrete chimney having an internal diameter of 8 ft. at the top and measuring 200 ft. high from the boiler-room floor line. At each boiler uptake there is a 5 x 8-ft. damper and a main damper in the smoke flue near the stack. Reviewing some of the dimensions, it will be seen that for each 10 sq.ft. of steam-making surface, there are 0.2 sq.ft. of connected grate, 0.1 sq.ft. of individual damper and 0.04 sq.ft. of stack area.

Ohio or West Virginia coal, the fuel burned, is dumped from wagons or trucks in the driveway directly into a 150-ton concrete bunker over the boiler furnaces. The bunker has a flat bottom, with gates opening to the magazines at each side of the stoker. Ashes from the furnaces are removed from the basement by an 8-in. steam-jet ash conveyor, provided with an intake in the bottom of each stoker pit. The ashes are discharged to a reinforced-concrete ash tank on the second floor and from there spouted to wagons or trucks in the driveway.

Boiler-feed water is heated in an open feed-water heater having 53.7 cu.ft. of steam space and 101.8 cu.ft. of water-storage space. The total volume of the heater is thus 165.5 cu.ft., which, in normal operation, calling for two boilers, reduces to approximately 0.2 cu.ft. per rated boiler horsepower. All condensation from live steam and exhaust lines is collected by pumps and traps and discharged to the feed-water heater, as well as the condensation obtained from the steam condensers and the heating system. Any makeup water required enters the heater under float control from the city mains.

Exhaust steam from the boiler-feed pumps and auxiliary steam pumps throughout the plant is used to heat the feed water, which is pumped to the boilers by either one of two 10- and 6- by 12-in. duplex piston-pattern pumps.

The new plant is a model of its kind, from both a sanitary and an operating standpoint. The complete undertaking, including buildings and mechanical equipment, was designed and installed under the direction of the Tait & Nordmeyer Engineering Co. of St. Louis.

Burton Duplex Water-Tube Boiler

T. Howard Burton, who for years has been indentified with the manufacture of a single-drum water-tube boiler of the old Worthington type, has perfected the two-drum duplex type of boiler shown in Figs. 1 and 2, and through the DePere-Burton Co., DePere, Wis., of which he is president, is placing this boiler on the market. It is a one-pass boiler for the gases provided with two cross-drums at opposite ends of the setting. Although the criss-cross arrangement of the tubes of the older boiler has been retained, the sectional construction has been eliminated. Instead of individual headers at either end of a tube section the lower ends of the tubes enter a large water-box header, extending the full width of the boiler. The tubes lead obliquely upward across the path of the gases and enter their respective drums. The header is thus eliminated at one end, and the arrangement gives maximum liberating surface for the steam generated, as each tube delivers its steam directly to the drum, thus eliminating the nipple connections between the header and drum common to most types of water-tube boilers. The water legs being arranged at right angles to the tubes, slope upward and outward at each end of the combustion chamber. An extension of the water leg with blowoff connections eliminates the necessity for a mud drum.

Water connection from the drum to the water leg on the same side and immediately beneath it is made by curved circulating tubes, one end leaving the drum radially and the other end entering the top of the water leg approximately in line with it. The curved tubes are easy to place and make a flexible connection, as they will bend slightly under the contraction and expansion of the boiler parts.

Fig. 1, a sectional elevation, shows how the tubes are arranged in vertical rows, adjacent rows sloping in opposing directions. A vertical row eleven tubes high leads from the header or water leg at the left side of

Fig. 1 and enters the drum at the right side of the drawing. In the next vertical row the tubes connect the header at the right-hand side of the setting and the drum at the opposite side. Thus the rows of tubes alternate in the direction of their slope and cross over the combustion chamber. This arrangement finally subdivides the gases of combustion, and as the space between the vertical rows of tubes is twice as great outside of the intersecting area, the gases—tending to follow the path of least resistance—spread out and cover the steam-making surface in the tubes. To further control the flow of gases, baffles may be placed horizontally, as indicated on the upper tubes.

As most of the tubes are curved, they allow for expansion and contraction and tend to reduce leakage that may be caused by movement at the joints. The tubes

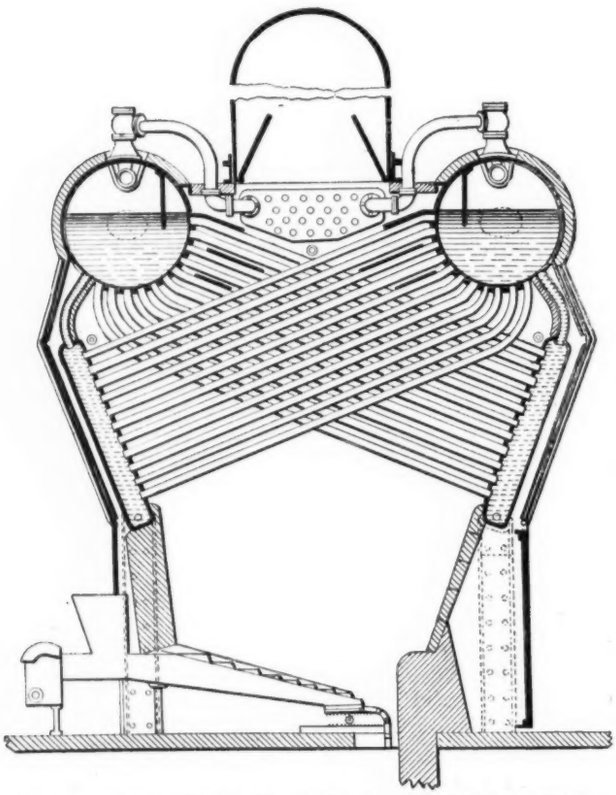

FIG. 1. LONGITUDINAL SECTION THROUGH BOILER

may be cleaned from within the drums and instead of pulling them out through handholes in the header, they may be removed in the combustion chamber. Both features are big time savers and tend to reduce the time it is necessary to keep a boiler off the line for cleaning or tube removal.

An outer shell is provided which may be varied considerably in construction and design. As will be noticed in the sectional view, about one-half of the surface of each drum is exposed to the heat of the gases. The short circulating tubes between the drums and the water legs are insulated from the hot gases so that the water in the drums may settle readily to the water legs and there will be no tendency to reverse the circulation.

This design of boiler can be installed in exceptionally low headroom, and the heating surface per square foot of floor space covered is unusually large. By the use of the large water-box headers, sectional headers and mud drums are eliminated. By leading the tubes into the drums one-half the header requirement is eliminated and maximum steam-liberating surface is provided. As a consequence the boiler steams rapidly. In the water legs key caps are provided opposite each tube, but the latter may be removed within the combustion space and

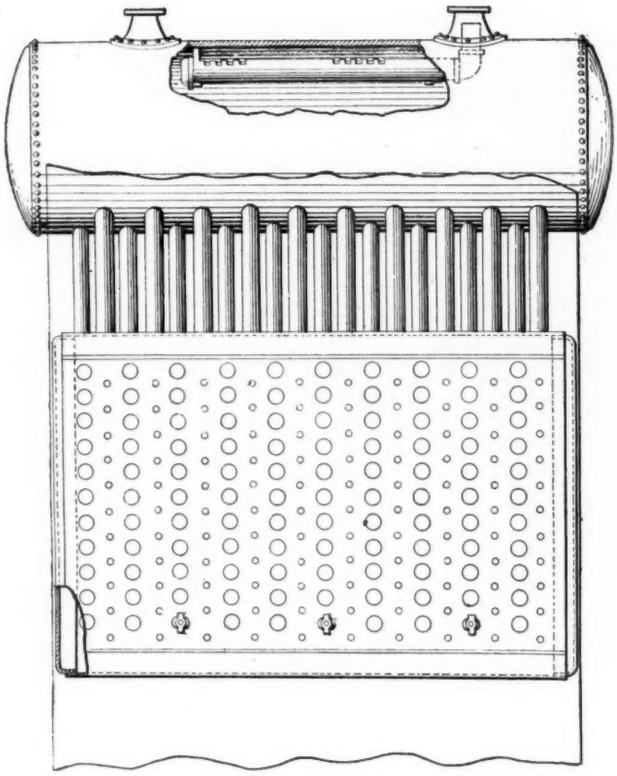

FIG. 2. CROSS-DRUM AND WATER-BOX HEADER

may be cleaned from the drum, so that the frequent removal of caps is avoided. The superheater is conveniently placed and being in the path of the outgoing gas, tends to insure a low exit temperature. Being centrally located and extending the full width of the setting, the uptake induces an even distribution of the gases over the heating surface. Three soot-blowing elements are provided, and as a result of their action the soot falls directly on the grate and is automatically removed with the ashes.

A paper entitled "Economical Supply of Electric Power," presented by William S. Murray at the recent convention of the A. I. E. E., has resulted in the appointment of the following special committee to be known as the "Committee on Super-Power System" to carry on the movement outlined by Mr. Murray: W. S. Murray, consulting engineer, chairman; Harold V. Bozell, of *Electric Railway Journal;* H. W. Buck, consulting engineer; John W. Lieb, vice-president, New York Edison Co.; Malcolm MacLaren, professor of Electrical Engineering, Princeton University; William McClellan, of Cleveland Illuminating Co.; Charles S. Ruffner, of North American Co., N. Y.; David B. Rushmore, General Electric Co.; Charles F. Scott, Yale University, and Percy N. Thomas, consulting engineer, New York.

Motors for Driving Ventilating Fans

Characteristics of Motors for Fan Drives—Comparisons Between Alternating-Current and Direct-Current Motors—Methods of Control—Formula for Calculating the Horsepower Required To Drive Fans.

By WILLIAM H. EASTON
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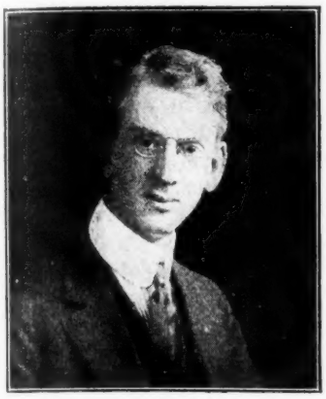


WILLIAM H. EASTON

VENTILATING fans are fairly simple devices, but unless they are driven by a motor that is properly selected to suit both the fan itself and the particular service for which it is used, the days of the man in charge will be full of trouble. There are many types of motor, each with its own special characteristics of speed, torque and mechanical construction, and there are several different methods of controlling the speed of each type of motor. However, only a few of these types are suitable for fan service under any circumstances, and when a particular application is carefully analyzed, it will generally be found that to obtain satisfactory and economical operation, the choice of proper motors is very limited.

The subject of the selection of motors for ventilating fan drives is a broad one and cannot be adequately covered in a short article, but there are some fundamental principles that may be of interest and assistance to operating engineers.

Reliability of operation is one of the first requisites of a satisfactory ventilating-fan motor, as the unit is generally placed in an out-of-the-way position, where constant supervision is impossible. Hence, the motor must be able to operate constantly with very little care and attention, which means superior commutation; dust-proof, nonleaking bearings with dependable lubrication; high-grade insulation, with protection against dust, moisture, acid fumes, etc.; and, especially, a record of satisfactory operation.

High efficiency of operation is also a necessary feature, for a ventilating-fan motor will ordinarily operate continuously for long periods of time, so that a difference of one or two per cent in the energy consumption will form quite an item in the annual operating expense. Thus, assuming energy at only one cent per kilowatt-hour, the cost of operating a 25-horsepower motor of 86.5 per cent efficiency at full load ten hours a day for 365 days will be $788, whereas a motor of 84 per cent efficiency will entail an expense of $810 for the same period. This shows that it would be poor economy to purchase a motor of low efficiency merely because of slight advantage in price, especially where the cost of energy is greater than one cent per kilowatt-hour. Efficiency must, of course, be considered for the speeds at which the motor is actually to run, and it will be found that some motors of good full-speed efficiency have a low efficiency at other speeds. Quiet operation is usually very important. Any noise made by the ventilating unit will be transmitted to other parts of the building through the air ducts, and any noticeable humming or roaring would be intolerable in such buildings as theaters, hotels, hospitals and offices. Mechanically, the motor should be light and compact, and when it is to drive an overhung fan, it must have ample strength of shaft and bearings to support the fan.

Either direct-current or alternating-current motors can be used for driving fans, and usually the question as to which type is to be selected is settled definitely by the kind of energy available. But where a choice is possible, it will almost invariably be made in favor of direct-current machines, for the following reasons:

Direct-current motors in all sizes can be obtained with slow speeds, so that they are suitable for direct connection to the ordinary type and size of ventilating fan. Alternating-current motors of the small sizes, on the other hand, are not supplied in slow speeds and hence must usually be belted to the fans they drive. There are some examples of alternating-current motors of less than 50 horsepower, which are directly connected to ventilating fans, but fans of such high speed are liable to be noisy in operation.

The speed control of direct-current motors is far superior to that of alternating-current types. Speeds that are both higher and lower than the normal motor speed can be obtained, and with proper selection of motor and control, efficient operation can be secured over the entire speed range. The speed of an alternating-current motor, however, can only be reduced below normal, increased speed being unobtainable; and on speed reductions below 20 per cent a considerable loss in efficiency occurs.

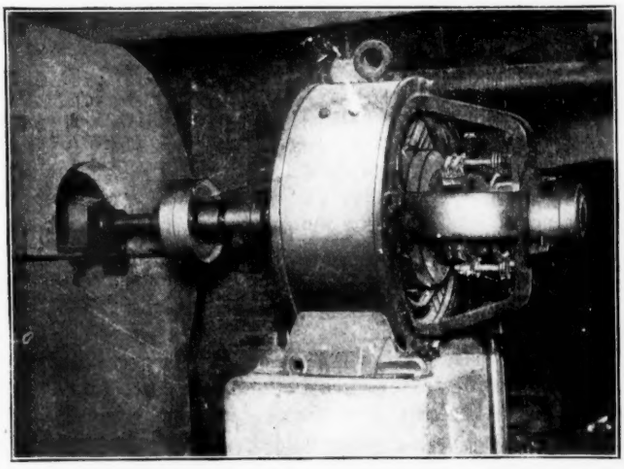


FIG. 1. DIRECT-CURRENT MOTOR DIRECTLY CONNECTED TO VENTILATING FAN

It must not, however, be assumed that alternating-current motors cannot be used for ventilating service. There are thousands of entirely satisfactory alternating-

current installations of all types throughout the country, and though in some extreme instances it may be desirable to install some means of producing direct-current, especially for the ventilating-fan motors, such cases are exceptional. Owing to the fact that there is an extensive demand for direct-current ventilating-fan motors, manufacturers have developed lines especially designed for this service. One such line comprises motors ranging from ½ to 50 horsepower in capacity, and almost all these are supplied in 27 different normal speeds, ranging from 90 to 525 r.p.m. These motors are of the shunt type for operation on 230-volt circuits and are designed for speed ratios of 1 to 2 by field control; that is, if the normal speed of the motor is, say, 300 r.p.m., it can be increased in any desired number of steps to 600 r.p.m.

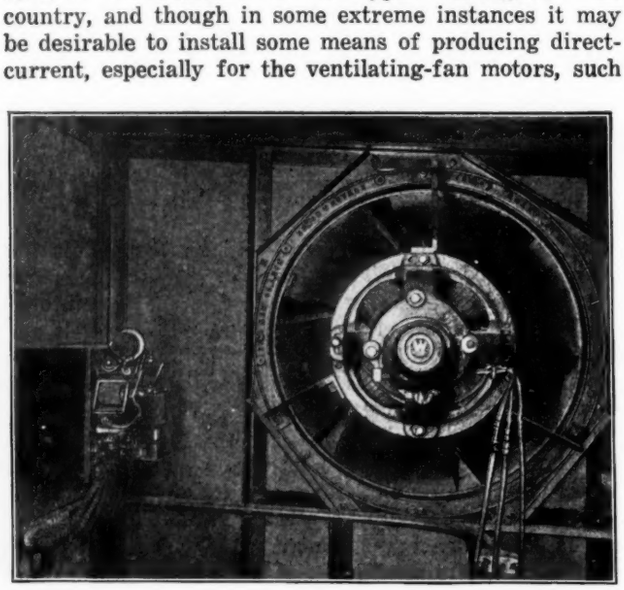


FIG. 2. FAN MOTOR CONTROLLED AUTOMATICALLY THROUGH PUSH-BUTTONS—CONTROLLER TO THE LEFT

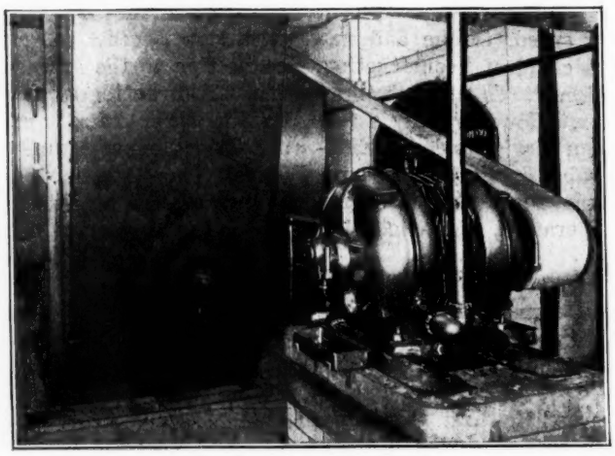


FIG. 3. SQUIRREL-CAGE ALTERNATING-CURRENT MOTOR BELTED TO VENTILATING FAN

For driving a fan continuously at constant speed, a motor of the proper normal speed is selected, and the control equipment consists simply of a starting box. Ordinarily, the starter is of the hand-operated type, but automatic starters are being used more and more extensively because the motor can be started and stopped from one or more push-button stations located where convenient. As ventilating fans are frequently installed in places difficult of access, remote control is often very convenient. Another advantage of the automatic starter is that it prevents injury to either the motor or the starter by cutting out the starting resistance either too slowly or too rapidly. If, as is usually the case, it is necessary to change the speed of the fan, either of three methods of changing the speed of the driving motor may be employed:

1. Field Control: In this method, resistance is introduced in the shunt-field circuit of the motor. The speed increases as the amount of resistance is increased, so that the normal speed of the motor is the lowest that can be obtained by this method of control. The chief feature of field control is that the loss of energy in the resistance is negligible. It is, therefore, used wherever possible, and where the speed range is not too great the general practice is to select a motor with a normal speed that corresponds to the lowest speed of the fan and to use field control for the higher speeds.

2. Armature Control: In this method resistance is introduced into the armature circuit of the motor, and as the effect is to reduce the motor speed, the normal speed of the motor is the highest that can be obtained by this scheme. Speed reductions of 50 per cent of the normal speed can be obtained by armature control, but as there is a considerable loss of energy in the resistance (amounting to about 19 per cent of the motor's input at full load), it is not often employed. However, where a fan is operated most of the time at constant speed, with occasional small speed reductions, this method is satisfactory.

3. Combined Field and Armature Control: Where a very wide speed range is essential, combined field and armature control can be employed, since with its use a standard ventilating motor can be operated over a speed range from one-half the normal speed to two times normal speed. Thus a motor with a normal speed of 300 r.p.m. would have a range of from 150 to 600 r.p.m., or a 1:4 ratio. Special controllers for all three methods are supplied by various manufacturers. Manually operated controllers are in general use, but automatic controllers with remote control are gaining in popularity. Automatic control usually consists of start-and-stop push-buttons with a manually operated speed-changing rheostat located on the man control panel, but it is also possible to control the speed by means of push-buttons.

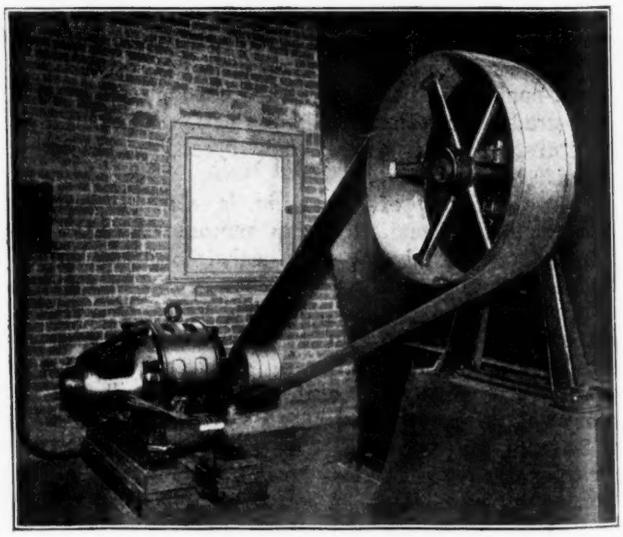


FIG. 4. WOUND-ROTOR ALTERNATING-CURRENT MOTOR BELTED TO VENTILATING FAN

There are two types of alternating-current polyphase

motors in general use for operating ventilating fans—the squirrel-cage and the wound-rotor. As stated previously, the smaller sizes are not made for slow speeds, so that they are ordinarily belted to the fans they drive, chain drive being too noisy for most applications. The squirrel-cage motor is the simplest type made, so that it is always used whenever possible. It is, however, strictly a one-speed machine, so that it can be used only for constant-speed applications.

The wound-rotor motor is similar to the squirrel-cage motor in construction except that resistance can be introduced into the rotor circuit. The use of resistance in the rotor circuit gives speed ranges similar to direct-current armature control; namely, it provides speeds less than normal, the minimum reduction being about 50 per cent, and as there is a loss of energy in the resistance, it is not particularly efficient except at small speed reductions. However, the wound-rotor motor will be found to be satisfactory for the great majority of ventilating applications, so that its use can be taken for granted in localities where alternating current only is available, unless a careful analysis of the application shows that a direct-current motor is necessary.

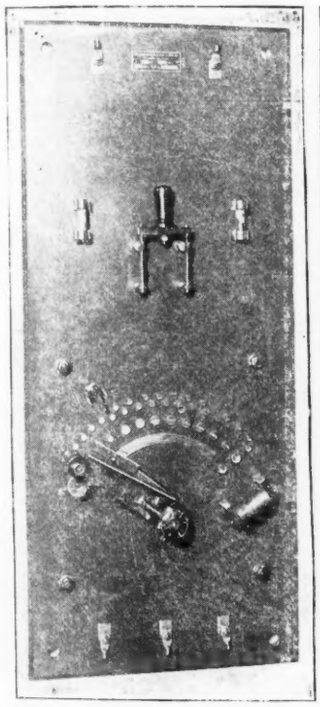


FIG. 5. MANUALLY OPERATED DIRECT-CURRENT STARTER AND SPEED REGULATOR

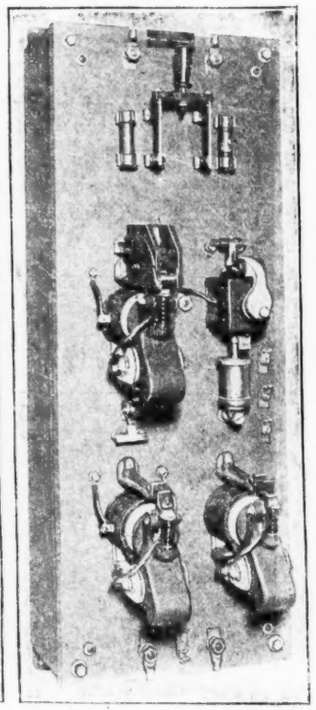


FIG. 6. AUTOMATIC DIRECT-CURRENT CONTROLLER FOR USE WITH PUSH-BUTTONS STATIONS

Wound-rotor motors are also frequently used for strictly constant-speed ventilating service because they can be started without taking excessive current from the line, whereas a squirrel-cage motor takes considerably more than full-load current during the starting period. Under certain conditions this characteristic makes the squirrel-cage motor objectionable, in which case wound-rotor motors are used instead. Alternating-current motors can be equipped with either hand-operated or automatic starters. Single-phase motors, in sizes up to 10 horsepower can be operated from lighting circuits and are extensively used for driving fans and blowers.

The horsepower required to drive a centrifugal fan can be estimated from the following formula:

$$HP = \frac{5.2 \times Q \times WG}{33{,}000 \times E}$$

Where

HP = horsepower of motor
Q = cubic feet of air per minute;
WG = pressure of air in inches by water gage;
E = efficiency of fan, which is approximately 0.5 for steel plate fan, 0.65 for Sirocco, and 0.45 for cone type.

The following mathematical relations are theoretically true for all centrifugal fans: Volume of air delivered varies as the speed; pressure of air delivered varies as the square of the speed; horsepower required varies as the cube of the speed. Hence a fan, which, when operated at 100 r.p.m. delivers 8,000 cu.ft. of air at ½ oz. pressure, requires 2 hp., will, when operated at 200 r.p.m., deliver 16,000 cu.ft. of air at 2 oz. pressure and will require 16 hp. It is therefore necessary, in selecting a motor for driving a fan, to figure the horsepower requirements at the maximum speed of the fan.

Reduction of the area of the inlet or discharge of a centrifugal fan will reduce the power required, whereas increasing the area of either will increase the power. Disc fans produce a relatively low pressure, and it is customary to figure their output on a basis of velocity. The relations between volume, speed and horsepower for centrifugal fans hold good for disc fans. However, if the outlet of a disc fan is restricted, the pressure and the power required increase, and with a totally closed outlet the power needed is almost double that with an unrestricted outlet.

Simple Belt-Lacing Rules

By E. J. Black

Improper belt lacing is probably the most common mistake that users make in the use of belting, and it is the one that causes the most serious trouble. By observing the following simple rules the belt will give the best service of which it is capable.

Cut the ends of the belt square. Do not depend upon your eye or use an ordinary ruler. If the end is cut slantwise, the pull will come on one side.

Make the holes as small as practicable. Use an awl rather than a punch, whenever possible.

Leave a sufficient margin at the edge of the belt without holes so as not to impair its strength. In belts 2 to 6 in. wide the holes should be at least ½ in. from the edge, ⅝ in. for belts 6 to 12 in. wide and ¾ in. for belts 12 to 18 in. wide.

Make two rows of holes in parallel lines straight across the width of the belt, and stagger them so that the strain comes upon different portions of the belt.

Be sure that the holes in the two ends to be joined match, otherwise there will be a "jog" in the belt, and this may result in tearing the belt lengthwise.

Use flexible lacing, being careful to have it proportionate to the size of the belt. A heavy lacing is likely to cause trouble.

In lacing the belt, make the pulley side as smooth as possible. Rough places and ends should be turned away from the pulley.

In using metal fasteners on rubber belts, select those which place the strain on the lengthwise strands. The crosswise strands are not so strong as those which run lengthwise.

Charts for Graphical Determination of Pipe Sizes and Velocity of Flow of Steam

By H. M. BRAYTON

Mechanical Engineer and Designer, Semet Solvay Company, Syracuse New York

A PREVIOUS article by the writer in the April 27 issue gave graphical charts for the determination of pipe sizes and steam velocities without calculation. The theory of the charts was given sufficiently to enable the reader to construct them himself and on any desired scale. The formulas given for the flow of steam involved the number of cubic feet of steam or quantity which passed a given section in a given unit of time. This quantity is not always desired, and if omitted the formulas may be combined and much simplified.

To recall these formulas they will be repeated here. Equation 1 gave the following:

$$Q = A\,V = 0.00545\,D^2V$$

in which

Q = Quantity in cubic feet per minute which passes a given section of pipe;

D = Diameter (internal) of pipe in inches;

V = Velocity of steam in pipe in feet per minute;

A = Internal area of pipe in square feet;

and equation 2 gave the following formula:

$$Q = W\,V'$$

in which

Q = Quantity as before;

W = Weight of steam flowing in pounds per minute;

V' = Specific volume of steam or number of cubic feet per pound. Chart for dry steam only.

It will at once be apparent that these two formulas may be combined and eliminate Q thus:

$$Q = 0.00545\,D^2V = W\,V'$$

in which the symbols designate the same values as before. This formula may be written,

$$\frac{0.00545\,D^2}{V'} = \frac{W}{V}$$

As this equation is a straight proportion, it may be built up into another form of Z chart. Such a chart would have the advantage that all reasonable values of the diameter may be given on one chart instead of two as was necessary when the value of Q was included.

Fig. 1 shows a chart drawn up for the foregoing formula. To use this, it is merely necessary to either know or assume four of the variables, and the other may be quickly found. It is more often desired to obtain the diameter of pipe necessary or the velocity of steam in a given size of pipe. In the illustration shown by means of dot-dash lines all the values are assumed except the weight of steam which may be expected to pass through the pipe. The values chosen are: Pressure, 230 lb.; pipe size, 12 in.; velocity of steam in pipe, 10,000 ft. per min. The pressure of 230 lb. represents a specific volume of about two as shown from the curve. This point is merely connected with the value of the diameter, and then the point representing the velocity on its scale is connected through the intersection of the first line with the diagonal and extended until it cuts out the value of the weight W on its scale. In a similar manner if the reader will connect the values of V = 12,000 and W = 30 found on their respective scales and mark the point of intersection of this line with the diagonal and then draw a line through this point on the diagonal to the value of D = 4 in., this line will be found to cut out a value on the V' scale of about 4.3. If this point is carried up to the pressure-volume curve and projected down to the pressure scale, it will be found to read 105 lb. This means that this pressure would be necessary to enable a 14-in. pipe to carry 30 lb. of dry steam at a velocity of 12,000 ft. per min. All this procedure is the work of but a few seconds and gives the results well within the desired range of accuracy.

The theory of this chart is somewhat different from those given in the previous article and will be given

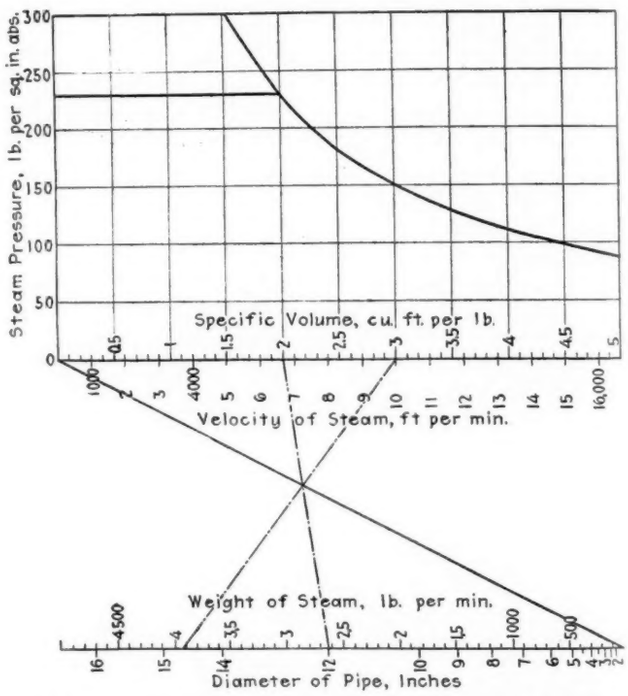


FIG. 1. CHART TO DETERMINE PIPE SIZE OF STEAM FLOW IN PIPES

here to enable the reader to construct it at his own convenience and at any size he may desire. Fig. 2 illustrates the method of procedure. Two parallel lines connected by a diagonal are drawn as shown. These lines may be any desired distance apart, as the theory is independent of the angle which the diagonal makes with the verticals. The chart is based on the simple laws of proportion, and these must be rigidly held to in determining the equations of the scales. Proportion is here illustrated by similar triangles which are formed by the dash-dot lines used in solving the formula by means of the chart. The equations of the scales follow:

$$X = m_1\,(0.00545\,D^2);\; Y = m_2\,V';\; Z = m_3\,W;\; T = m_4\,V$$

The values of m_1, m_2, m_3 and m_4 are known as the modulus of the respective scales and may be of any value

the designer desires. Their actual numerical value is determined on the limits of the variables and the size of the chart which it is desired to construct. These two points must be settled upon before actual construction can begin. The limits which the writer chose in this case are shown on the chart, Fig. 1. In general the greater the limitations desired in the chart the better accuracy obtainable. If the reader never has occasion to deal with steam pipes larger than 10 in., for example, the chart should not be made for larger size.

Once the limits are determined and the length of the various scales settled, it is a simple matter to determine the values of the moduli. It will usually be found convenient to make the scales about 10 in. long. This

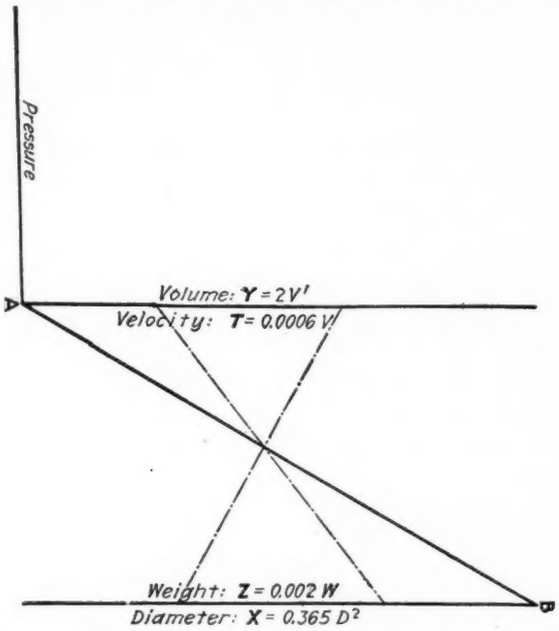


FIG. 2. ONE OF THE STAGES OF CONSTRUCTING THE CHART SHOWN IN FIG. 1

10 in. will then represent the value of X, for example, when D has its maximum value. With X equal to 10 and the maximum D known, it is simple arithmetic to solve for m_1. The other moduli are found in the same manner except m_4. When the other three have been determined, then m_4 is fixed because from the nature of the proportion and similar triangles the following must hold:

$$\frac{m_1}{m_2} = \frac{m_3}{m_4} \text{ or } m_4 = \frac{m_2\ m_3}{m_1}$$

It is, then, necessary only to select the first three moduli and the fourth follows automatically. Of course it is possible to definitely assign values to any three of these moduli and calculate the fourth. The reader must not assume that only the first three can be determined by the designer. In fact, it often happens that it is better to select the others. The designer has a free hand in this matter. Again, it may be found that the fourth scale calculated comes absurd; that is, much too small or too large in proportion to the other scales. In such a case the designer may be able to use a multiplier in the proportion. He may multiply the numerators or the denominators of the fractions by some number. Of course it is evident that if the numerator of one fraction is multiplied by a number in this manner, the other must be also; otherwise the proportion would not hold and the formula would no longer be true. In the formula being considered it would be permissible to multiply the V' by a number if the value of V were also multiplied by the same number. Simply multiplying the two terms of a fraction by a number does not affect the results. In some cases, when the moduli do not work out well, it may be necessary to change the limits of the variables or the length of the scales.

In the chart shown here the following moduli were selected:

$m_1 = 6.67$, $m_2 = 2$, $m_3 = 0.002$, $m_4 = 0.0006$. It is always advisable to select these values in even numbers when possible. By this is meant not necessarily even digits, but numbers that are easy to lay out, such as the value of m_3 and m_4 given above. This is not always possible, but should be kept in mind and will save time in constructing the chart later on. The principal point in selecting these values is to make all the scales take all the room allowable. It will be observed in this chart that all the four scales are of about equal length and take the full length of the line. The equation of the scales in our problem then becomes,

$$X = 6.67\ (0.00545D^2) = 0.0365D_4 = Y\ 2\ V';$$
$$Z = 0.002\ W;\ T = 0.0006\ V.$$

With these equations it is a simple matter to lay out the scales. With different values of the variables substituted into their respective equation, a corresponding value of X, Y, etc., is obtained. These values are distances and are to be laid off from the point of intersection with the diagonal. Each point must be marked with the value of the variable used in obtaining the distance. For example, in the V' equation when $V' = 2$, then $Y = 4$ in., which means that a point 4 in. from the starting point is marked and labeled 2 cu.ft. on the diagram.

The pressure-volume curve is plotted from values taken from the steam tables for dry or saturated steam. This curve is plotted to enable one to quickly determine the pressure without reference to the tables.

British Control of Fuel Oil

In a recent address, Paul Foley, Director of Operations, United States Shipping Board, stated that although 65 per cent of the world's current petroleum production is being drawn from the United States, practically all the visible future production is under the control of Great Britain. Furthermore, although production in the United States is available to all other nations on equal terms with our own, that under British control and located in the middle area of the world is available only to British nationals. The practical effect of these restrictions is that while British ships can secure fuel oil in United States and Caribbean ports on the same terms as our own and also obtain what they require overseas from the nearest producing field, American requirements overseas can be met only at British terms and must reflect the long haul from American and Mexican seaboards. An important step in correcting this evil is that we should take advantage of the fact that the control which our British cousins enjoy has been attained by means of capital loaned by the United States Government. With this in mind we should insist that the same principle of equal opportunity now extended to British nationals in American producing fields be extended to American nationals in British producing fields.

Overhauling and Aligning the Engine

Detailed Instructions Regarding the Alignment of the Connecting Rod, the Crosshead Guides, Crosshead-Shoe Adjustment, Vertical and Horizontal Alignment of the Crankshaft, Worm Bearings and Crankpin Error

By THOMAS W. AIREY

IT IS obvious that in the course of time the power-plant engine will require overhauling and repairing, and all repairs must be skillfully executed and the final adjustments made in accordance with the alignment of the various parts. The want of alignment may have been made manifest by the constant heating of the main bearing, the heating or pounding of the crankpin brasses or by a knocking in the cylinder and, possibly, by an appreciable loss of power. As a general rule this want of alignment is induced by wear or is incurred by repairs or adjustments that are made from time to time without due consideration being given to the alignment.

An error of alignment may exist in any direction, but the error may be located, if the test is made at four equidistant parts of the revolution; for instance, on the two dead centers of the crank and at the highest and lowest points, or half-stroke of the path of rotation of the crankpin.

It is not an uncommon practice for the engineer to use the connecting rod as a means of locating an error of alignment in any of these directions because, owing to the length of the rod, the error becomes magnified and is thus more plainly discernible, but in using the connecting rod as a means of testing the alignment one must know that the bore of the crank brasses are at a true right angle with the center line of the rod.

Aligning the Guides

A not uncommon method employed by engineers for aligning the guide with the cylinder and main bearing is by using the piston and gland after the rings have been removed from the piston and the packing from the stuffing box. The bottom shoe of the crosshead is then adjusted until the gland enters the stuffing box freely when tried from any part of the stroke. Suppose that the piston is a good fit in the cylinder and that the gland is a loose fit on the rod; then the piston rod would fall in the direction of the line *AB*, Fig. 1, *AA* representing a line through the center of the cylinder bore and in alignment with the center of the main bearing *C*. Then it follows that if the guide is set by the gland only, it would be set to the line *B* and would, therefore, be out of alignment with the main bearing, as shown.

Suppose that the gland was a good fit to the piston rod and stuffing box, but that the piston was a slack fit in the cylinder; then the gland would not remain tight because the piston, being suspended in the cylinder, the overhanging weight of the piston would cause rapid wear of the neck ring, the piston-gland bore and also of the guide surface, and in a direction parallel to the line *AE*, Fig. 2, thus gradually letting the head of the piston fall to the bottom of the cylinder bore, touching the end of the cylinder first at *F;* thus the close fit of the piston in the cylinder and the gland to the piston rod and stuffing box are alone depended upon as a guide whereby to set the piston rod and its guide in alignment with the center of the main bearing.

The better method of aligning is that of stretching a line through the cylinder and stuffing box, the line being passed through a small hole in the end of a piece of iron plate *G*, Fig. 3, that is secured to the head end of the cylinder by one of the stud bolts. The line is then passed through the bore of the stuffing box and over a supporting bracket *H*, a weight *I* being attached at the end to keep the line tight. The line *AA* is then set central to the bore of the cylinder and stuffing box, as shown.

To test the guide path for alignment (supposing the guide to be of the channel pattern), take a height gage *G*, Fig. 3, and placing the base on the surface of the path *J*, adjust the point of the gage until it just touches the line at *K*. Then move the gage to the other end of the guide and test the distance *L*, and if *K* and *L* are coincident, then the guide is in alignment. If *K* and *L* are not coincident, the guide must be scraped down until the two measurements coincide, and finally bedded to a surface plate. The top surface of the guide must next be tested for parallelism with the bottom surface and, if incorrect, must be treated in a similar manner.

Next make a gage *M*, Figs. 4 and 5, having a true and straight edge and fit it in between the flanges of the path at one end, as at *N* and *O*, Fig. 4. Then take a square *P*, and while pressing the back firmly upon the guide surface, bring the blade gently up to touch the line *A*, holding the square firmly in this position, and with a fine-pointed scriber draw the line *Q* on the gage *M*, using the blade of the square as a guide. Then move the gage to the other end of the path and, with the square held firmly on the guide path surface, test the line *Q*, already marked on the gage, with the line *A*. If the two coincide, then the inside faces of the path flange are true and parallel to the line *A*. Should, however, the width between the inside faces of the guide flange, when tried at each end of the gage, not coincide, then the inside faces must be made true and parallel with the line *A*.

Testing the Crosshead

Take the crosshead *R* and fit a true taper mandrel in the piston-rod bore and then place the mandrel between lathe centers, as shown at Fig. 6. Then adjust the crosshead in the lathe centers until the side face of the crosshead is true to a square tried from the lathe bed. With a scribing block, the point being set to the lathe-spindle center and the base resting on the lathe bed, draw a line across the boss end of the crosshead and also the gudgeon end of the crosshead, as shown at *S* and *T*. Then move the crosshead round in the centers one-fourth turn, as shown in Fig. 7. Setting the sliding surface of the crosshead true with the square from the lathe bed and with the same setting of the scribing block, draw a line across the ends of the crosshead shoe, as shown at *U V*, and mark the line lightly with the center punch, taking the measurement from the

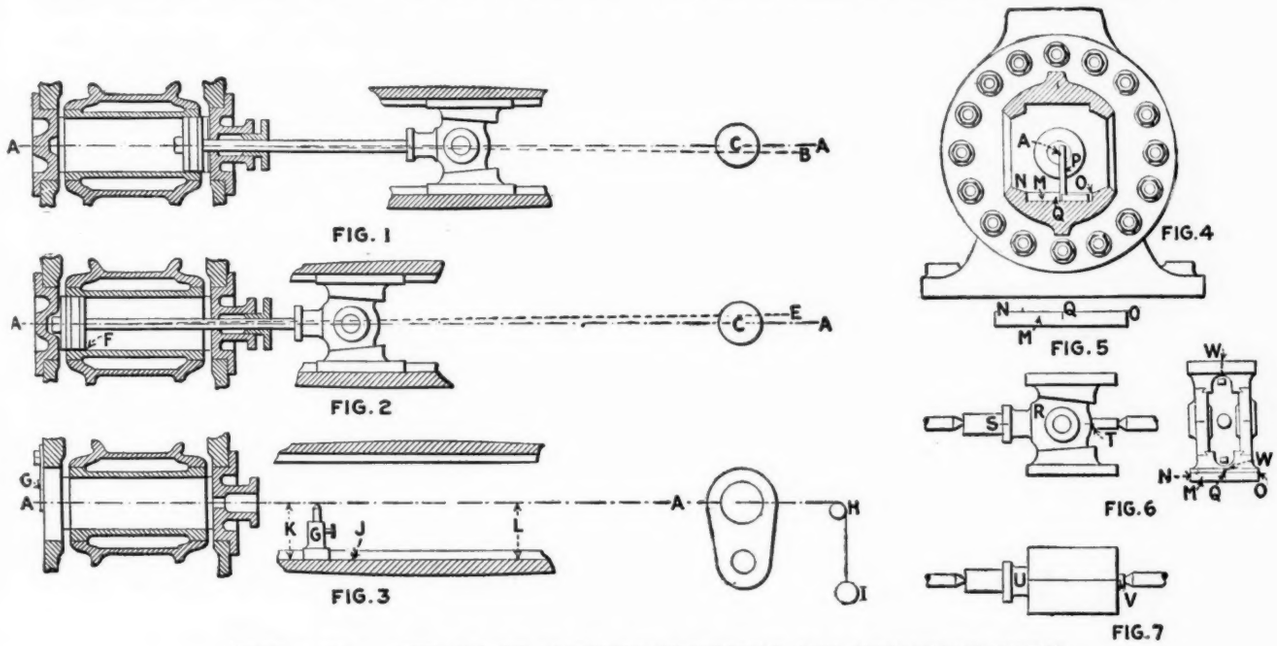
gage *M*, Fig. 5. Check them with the crosshead or take the gage *M* and lay it on the end of the crosshead shoe, as shown by Fig. 6, the point *Q* being set to coincide with the lines *WW* on the shoe. If the crosshead is true, the ends of the gage *N* and *O* should be coincident with the face of the crosshead shoe. If, however, the two are not coincident, and it should be found necessary to provide new shoes, the measurements must be performed in strict accordance with the marking *NQO* on the gage *M*. The top of the path and the top shoe of the crosshead must also be tested in like manner.

The bottom shoe of the crosshead may next be adjusted until the distance from the sliding surface of the shoe to the center line *ST*, Fig. 6, and the distance from the surface of the guide path to the line *A*, as shown at *K*, Fig. 3, are coincident. The bottom shoe must also be tested for parallelism with the center line *ST*. The crosshead is then ready for bedding in the guide, the top shoe being adjusted to the desired fit.

To test the vertical alignment of the crankshaft, or to ascertain if the crank is parallel with the crosshead journal, move the crank *W*, Fig. 8, around to nearly the dead center or until the crankpin *XH* just touches the line *AA* that is central through the cylinder, as shown. With a pair of calipers test the distance *YA* with that of *ZA*, Fig. 9. Then move the crank around until the pin touches the line *A* on the other dead center and test the distance *AB* and *AC*. If all measurements coincide, the crankshaft is in true alignment. If, however, *ZA* and *CA* are equal, but measure less than *YA* and *BA*, the center of the crankpin journal length will be outside the line *AA*. This may be caused by the flange of the main-bearing brass being too thick, if a new one has been put in.

Supposing that *YA* and *BA*, Fig. 9, are not coincident and that the crankshaft, instead of being in alignment, as represented by the line *D*, Fig. 10, is in the position represented by the lines *E*. Then the center of the crankpin journal would be on one side of the line *A*, as at *F*, on one dead center, and on the opposite side of the line *A*, as at *G*, when tried on the other dead center. If the connecting rod were coupled up to its bearings under such conditions, the crankpin journal would heat, and if any end play existed there would also be a knocking or pounding. In order to bring the crankshaft into true alignment, a side adjustment of the main-bearing brasses must be made until the line *AA* cuts the center of the crankpin journal when tried on both dead centers.

If the connecting rod is used to test the vertical alignment and it is coupled up to the crankpin, it would be held parallel with the line *H;* thus the other end of the rod would be thrown over to one side of the crosshead pin. Then move the crank round to the other dead center, and the small end of the rod would still be thrown over to the same side of the crosshead pin, but to a lesser degree. Thus, by testing the alignment with the crank on the dead center nearest the cylinder, the error is more plainly indicated.



FIGS. 1 TO 7. DETAILS OF ALIGNING CROSSHEAD AND CROSSHEAD GUIDES

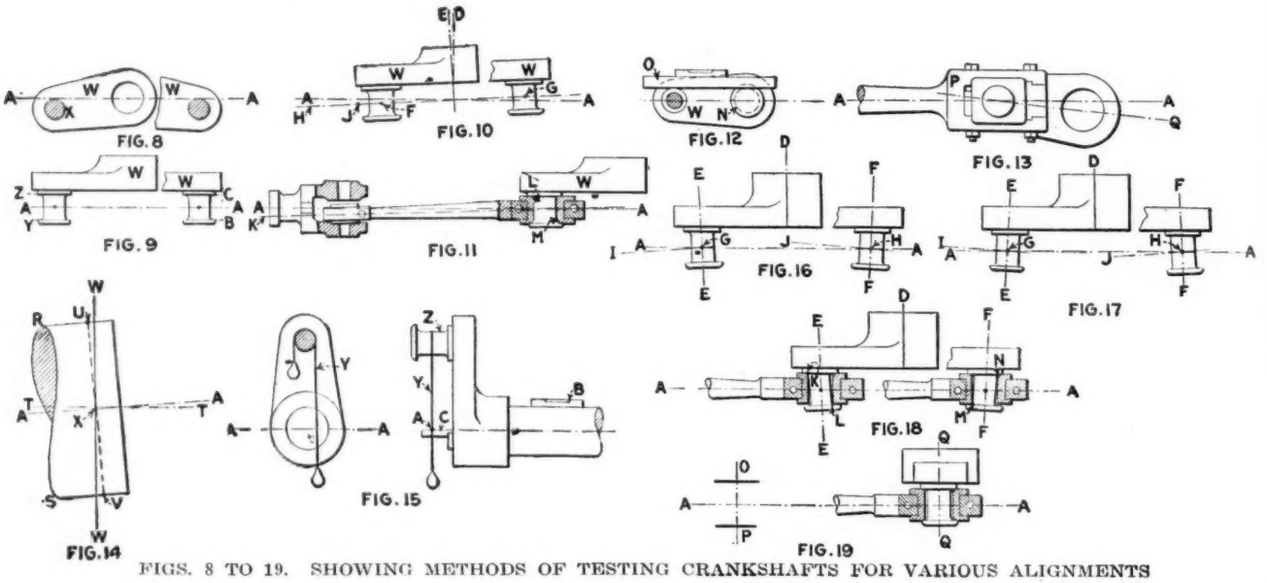
The reason for this will be seen by referring to Fig. 10, where *AA* represents a line true with the cylinder bore and the guide; *E* is the center line of the crankshaft, and *F* and *G* represent points central to the length and diameter of the crankpin. The center line of the connecting rod is represented for one dead center by a line passing through *H* and *F* and for the other dead center by *JG*. It will also be observed that the point *J* falls on one side of the line *AA* and that the point *G* falls on the opposite side, although the center line of the connecting rod is the same in both cases cited.

It is not always wise to couple up the connecting rod to its respective bearings even though it is known that the crankpin is in true alignment unless one is assured that the bore of the large and small-end brasses are at a true right angle to the center line of the rod. Even then it is better to take no risks, especially as an error of $\frac{1}{64}$ in. in a 3-in. length of journal will throw the rod out of alignment $\frac{3}{8}$ in. in a 6-ft. length.

Couple the large end of the rod to the crankpin and tighten up the adjusting screws so as to just be able to move the rod. Then try the other end of the rod with the crosshead, and if the rod points true with the jaws of the crosshead, it is in alignment. If the rod,

instead of being true with the crosshead, points over to one side, as represented by the line *K*, Fig. 11, the bore of the crankpin brasses must be scraped out at *L* and *M* and rebedded until the rod points true in the crosshead, or parallel with the line *AA*. The rod should also be tried for alignment with the crankpin by coupling up the small end of the rod and lowering the other end onto the crankpin. When it is true, it will fall laterally between the collars of the crankpin.

After making repairs or adjustments, it is a good plan to check up the piston clearances, for if the clearances are kept right no ridge should form at the ends overrun a little into the counterbore, but it is possible of the cylinder or guide, as part of the piston head will that a small ridge may form at the cylinder ends due to the weight of the piston head and rings, which pass but part way over the end surface of the cylinder. It therefore becomes obvious that the connecting rod should be kept at its original length, as near as possible, by using suitable liners with the brasses. If this is neglected, a ridge may form in time in the cylinder, and if due consideration is not given to the changing length of the rod it may become so adjusted as to cause the piston head to strike the ridge, a knock in the cylinder being the result.



FIGS. 8 TO 19. SHOWING METHODS OF TESTING CRANKSHAFTS FOR VARIOUS ALIGNMENTS

A method of testing the horizontal alignment of the crankshaft, viewed from the crankpin end of the engine, as adopted by some engineers, is to take a spirit level and place it across the guide path or on some machined part on the top of the cylinder and note the position of the bubble. The level is then placed along the top of the crankshaft, and if it indicates that the shaft is wearing low in one of its bearings, the main bearing brass is set up until the bubble in the level coincides with the position when tested on the cylinder top. Another method sometimes adopted for testing the horizontal alignment is to find the center at the end of the crankshaft, as shown at Fig. 12. From the center scribe the dotted circle *N* of a diameter equal to that of the crankpin shoulder. If there is no crankpin shoulder, then the circle must equal the diameter of the crankpin. The crankpin is then placed on the dead center and a straight edge *O* placed with the edge coincident with the shoulder of the crankpin and the perimeter of the circle *N*. The spirit level is then placed on the straight edge, as shown, and the main bearing is raised until the bubble stands in the same position as when the level is placed on the cylinder bore or along the piston rod or guide path.

Supposing that the main bearing has worn low, this would cause the crankshaft to drop from its normal position, which is in line with the center line of the cylinder guides and connecting rod, to below the center line; then the crank would be out of alignment as represented by the line *P Q*, Fig. 13. Should the wear of the brasses be excessive, the result would be serious, as the crank, dropping low in its bearing, would cause part of the piston movement and pressure that should be exerted on the crank after leaving the dead center to be exerted on it before it reaches the dead center, thus causing a back pressure and involving a loss of power.

If the connecting rod is used as a means of testing the horizontal alignment of the crankshaft, the test should be made with the crank in its highest or lowest, or half-stroke position. Should the test be made with the crank on the dead center, assuming that the vertical alignment of the crankshaft is correct, then the connecting rod end would fall laterally between the crankpin shoulders; whereas, if the test were made with the crank on its highest and lowest position, or half-stroke, and the bearing was wearing low, the error would be plainly seen, the connecting rod falling on one side of the crankpin when in its highest position and on the other side when tried with the crank in its lowest position.

This will be more clearly understood by referring to Fig. 14, where *AA* represents a line through the crankshaft, which is somewhat out of parallel with the crosshead journal, owing to the wearing down of one of its bearings. *R* and *S* are lines representing the crankpin in its highest and lowest positions, and *T* represents a line through the crosshead pin which is in true alignment with the cylinder and guide. *U* and *V* are points central to the diameter and length of the crankpin, and *WW* denotes what would be the path of revolution of the crankpin were the crankshaft in true alignment with the line *TT;* but as the shaft is out of alignment, the path of revolution is from *U* to *V*, the center of the crankpin falling on one side of the line *WW*, as at *U*,

in its highest position, and on the opposite side, as at *V*, at its lowest position. Suppose that the connecting rod is coupled up and adjusted to the crankpin *R*, then the bore of the crankpin brasses would be held in a position parallel to *R*, and the bore of the small end would be held parallel to *TT*. When the crank is in its lowest position, the bore of the crankpin brasses will be held parallel to *S* and the small end will be held parallel to *TT*. Starting from the dead center, *X*, the twisting of the rod will gradually increase, attaining its maximum when the crank reaches its highest point, or half-stroke, and then gradually diminishing until the other dead center is reached, when on passing over, the twist will again be taken up, attaining its maximum when the crank reached its bottom or half-stroke position, as shown at *V*, and again gradually diminishing until the dead center is again reached.

If the connecting rod is used as a means whereby to test the horizontal alignment of the crankshaft, the test must be made with the crank in its highest and lowest, or half-stroke position, for reasons explained in reference to Fig. 14. To test the horizontal alignment of the crankshaft by a line *AA*, Fig. 15, through the cylinder and guide path, use a plumbline hung over the crankpin and adjusted to a position central to the crankpin length, as at *Z*. If the shaft is in alignment, the plumbline should just touch the line *AA*, but if otherwise a spirit level is placed along the top of the crankshaft, as shown at *B*, and the main bearing is adjusted until the bubble stands in the center and the plumbline *Y* just touches the line *A*. If the center of the crankshaft and the line *A* are coincident with a square *C* held against the face end of the crankshaft, the edge of the blade should coincide with the shaft center and the line *A*, as shown.

Testing Alignment of Crankpin

Testing for crankpin error in alignment with the crankshaft is more difficult. It has already been pointed out that an error in alignment may exist in any direction, but if the test is made at four equidistant parts of the stroke, as, for instance, on the two dead centers and at the highest and lowest or half-stroke position of the crank, the error may be located. If the connecting rod is coupled to the crankpin, the crank being placed on the dead center, and the small end of the connecting rod is uncoupled, it will be difficult to determine, should the small end be out of alignment, whether the error is due to want or alignment of the crankpin or to the crankshaft itself. However, there is this difference between the two cases. If the error is due to want of alignment of the crankshaft, the small end of the rod will point out on the same side of the crosshead pin no matter on which dead center the crankpin stands; but when it is due to the crankpin being out of alignment, the rod end will fall on one side of the crosshead pin on one dead center and on the opposite side when tried with the crank on the other dead center.

It has also been shown that when the shaft is out of alignment, the center point of the crankpin would fall on one side of the cylinder center line *AA* on one dead center and on the opposite side when on the other dead center, as at *F* and *G*, Fig. 10, and at *U* and *V*, Fig. 14; but when the crankpin is out of alignment with the crankshaft, the path of the center point on the pin will remain in the true plane.

This is shown at Fig. 16, where *D* represents a line through the crankshaft; *E E* represents the center through the crankpin; *F F* represents a line through the center of the crankpin when on the other dead center; *G* and *H* are points central to the crankpin on the respective dead centers; and *A A* is the cylinder center line and also represents the path of rotation of the crankpin center point. Thus, during the path of rotation of the center points the crankpin remains in the true plane.

Crankpin Out of Alignment

Suppose that the crankpin is out of alignment in the direction shown in the figure and the connecting rod is coupled up to the crankpin; then the center line of the connecting rod would be held in the position represented by the line *I* and the small end of the rod, being coupled up to the crosshead pin, which is at right angle to *A A*, the rod at that end is held in a position parallel to *AA*. Then the adjusting of the brasses to their respective journals would tend to bind or spring the rod, one end being held parallel to *A A* and the other parallel to *I*. On the crank reaching the opposite dead center, *FF* representing its center line and *J* the center line of the connecting rod at right angle to *FF*, the want of alignment in the pin would throw the small end of the connecting rod on the opposite side of the crosshead on the cylinder center line *AA*, as represented by the line *J*. Thus the connecting rod is sprung twice at each revolution and is twice jammed from side to side of the crosshead at each revolution.

Suppose the error in the alignment of the crankpin is shown to exist in an opposite direction, as in Fig. 17, *EE* representing a line through the crankpin on one dead center. *AA* is a line true to the cylinder bore, *D* is the center line of the crankshaft, and *G* and *H* are points central to the diameter and length of the crankpin. It will be obvious that the position of the connecting rod will be reversed from that shown by Fig. 16, and that in coupling the connecting rod to the crankpin the rod would be held in the position of the line *I* and the rod end will be thrown over to the opposite side of the crosshead or cylinder center line *AA*, as shown. As the crankpin reaches the other dead center, the rod will be held in the position parallel with the line *J*. It is by this difference that it is possible to know in which direction the crankpin is out of alignment.

Supposing that the crankpin is out of alignment when the crank is in its top and bottom or half-stroke position, then with the connecting rod coupled to the crankpin it would fall to one side of the crosshead pin with the crank on its top half-stroke, and to the other side when tried with the crank on the bottom half-stroke. Thus again the direction in which the crankpin is out of alignment may be known by its position.

The result of an error in the crankpin alignment is shown by Fig. 18, where the large end of the connecting rod is coupled up to the crankpin. If the connecting rod is coupled up at the small end to the crosshead pin, which is in true alignment, the connecting rod will be held true and parallel to *AA*. It will be seen that the bore of the crank brasses come in contact with the crankpin only at the points *K* and *L* on one dead center and at *M* and *N* on the other dead center. If the crank is moved around, bringing the crankpin at the top of its revolution, as shown in Fig. 19, the center line of the pin will stand parallel to that of the crosshead pin, *O P* representing the crosshead pin and *Q Q* a line through the crankpin.

Play will therefore exist between the pin and the bore of the brasses. Starting from the dead center, as shown by Fig. 18, the play or lost motion gradually increases, attaining its maximum when the pin reaches the half-stroke position, as shown by Fig. 19. On passing the half-stroke position the lost motion gradually diminishes until passing the dead center, when the rod is thrown over to the side of the crosshead pin and at the same time the crankpin pounds in the connecting-rod brasses.

On the crankpin reaching its bottom half-stroke position, the lost motion has again reached its maximum and again diminishes as it reaches the other dead center, and on passing over the rod is again forced to one side of the crosshead pin, but in an opposite direction, and at the same time a pounding of the crankpin in the connecting-rod brasses again occurs. Thus the rod is thrown over to the side of the crosshead twice in each revolution, and the crankpin pounds twice in each revolution. If an attempt is made to stop the pounding by tightening up the brasses to take up the lost motion, the end will be sprung or twisted and heating will result.

Improving an Inefficient Water Power-Plant

By John S. Carpenter

A few changes in the shape of the flume resulted in increasing the output of a 66-in. vertical hydraulic turbine, from about 92 per cent to 107 per cent of its rating.

This turbine, which was direct-connected to an umbrella-type generator on the floor above, was rated at 600 hp. under 15 ft. head. The owner complained that he could only get 554 hp. out of the unit with the gates wide open. The waterwheel builder disclaimed all responsibility when he found that the

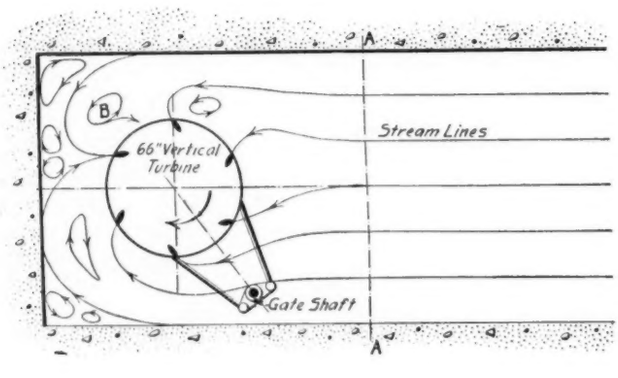


FIG. 1. ORIGINAL SHAPE OF FLUME, SHOWING STREAM LINES

true head acting on the wheel was only 14.5 ft. as against 15 ft. and left the owner to his own devices. Power, the owner said, was worth at least $150 per hp. per year and he could afford to spend up to $5,000 to get the plant running so that the wheel would give its rated power.

After the plant was thoroughly gone over, among other things it was found that the water velocity at a cross-section marked *AA* in Fig. 1 was about 3.1 ft. per sec. at full gate. This velocity is quite high, since good practice would limit the velocity at such point to 2 ft. per sec. The square corners in Fig. 1 are not favorable to good efficiency, and at the point marked *B* there was a distinct pocket about 10 in. deep and about 12 in. square in plan. This was credited to the high velocity and the reversal in the stream at that point. Further trouble from eddies being suspected, the owner was asked to dump a quantity of

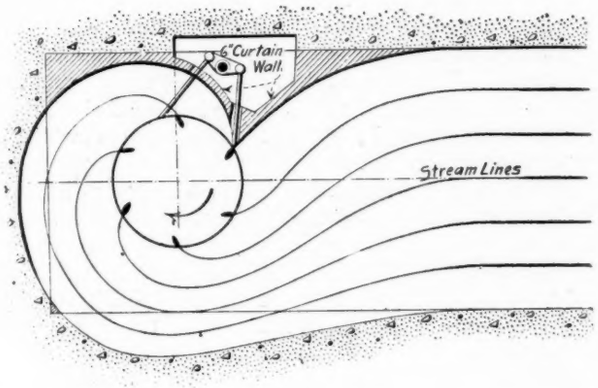


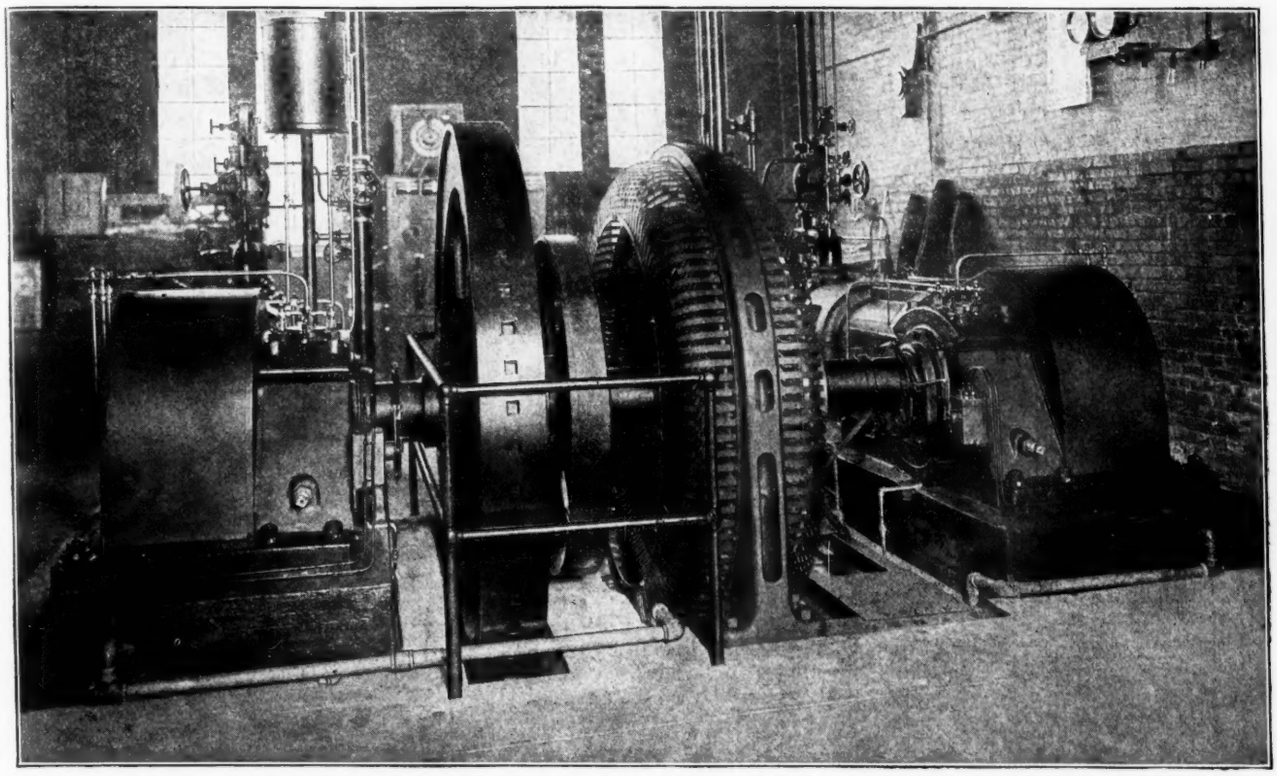
FIG. 2. SHAPE OF FLUME AS ALTERED, SHOWING IMPROVEMENT IN STREAM LINES

shavings and thin sticks into the penstock and take several flashlights of the portion above the wheel. The stream lines shown in Fig. 1 were traced from the resulting photographs of the direction and distribution of the sticks.

It is to be regretted that the actual photographs cannot be reproduced on account of their being lost, negatives and all. The lines of flow were rather well defined back to a point behind the gateshaft; beyond that it was difficult to trace out the paths, because the water flowed more or less uniformly. It should be mentioned that to get the effect shown it was necessary to close down the head gates about halfway to lower the headwater. It will be seen that the gateshaft was put in about the worst place possible, as far as efficiency goes. The reversals and eddies are very plainly shown on the original photographs and caused quite a loss in power.

To straighten out matters, the writer recommended that the flume be made spiral in form as shown in Fig. 2. This meant cutting into a concrete wall as much as 30 in. in one place, replacing the reinforcing steel and building a projection on the outside of the wall. The proposition was looked on with anything but favor. Finally, it was decided to cut right through the wall and replace with a new section. While the tearing up was going on, it was found that the gateshaft could be moved to the opposite side of the penstock out of the waterway, and behind a thin curtain wall.

It is not alleged that the outline shown in Fig. 2, a tracing of an actual photograph, is the best possible shape. It is merely a patch job. As to results, by changing the shape as shown and by doing some tailrace excavation, whereby 9 inches more head was obtained, the output was increased to 639 hp., to the joy of the owner. The total cost of the changes was $2,786, and the work was done in the dry season, when water was low and scarce.



AMMONIA COMPRESSOR OF THE HIGH-SPEED TYPE, ELECTRICALLY DRIVEN, AMERICAN ICE CO., WASHINGTON, D. C.

Refrigeration Study Course—IX
The High-Speed Ammonia Compressor

By H. J. Macintire

As ALL engineers know, the four-valve steam engine, and particularly the Corliss type, is the most efficient of all reciprocating engines, And yet there are many medium- and high-speed engines that have a ready sale for all kinds of work and to all locations. The reasons for this substitution lie in the fact of smaller size and smaller first cost of the machine itself. It is not expected that so high a prime-mover efficiency will be obtained, but these considerations, as regards lesser floor space required, lesser engine-room volume, possible use of the exhaust steam for heating purposes, and lower initial cost of the installation, overbalance the disadvantages.

In the refrigerating field, however, the conditions are not exactly the same. A score of years ago rotative speeds of 30 to 40 r.p.m. were usual, then came speeds of 60 to 70 r.p.m., and finally 100 r.p.m. for the larger and 140 to 160 for the smaller machines (usually the twin vertical type). Of late years attempts have been made to increase the speeds still more, and encouraging success has been obtained, the main difficulties encountered being purely mechanical ones due to poor lubrication or faulty machine design. Defects which cause wrecking of the machine and loss of life are infrequent. Designed properly, so as to maintain lubrication and to be amply strong, the future of the high-speed compressor seems bright, especially as the high speeds make it possible to connect directly with synchronous motors in sizes of fifty tons or over. As will be seen later, there is no appreciable loss in economy at high speeds provided a good form of valve is used and large opening area for the valve provided.

As has been shown in another part of this study course, the refrigeration duty is dependent on the weight of ammonia pumped and condensed in the condenser. For any particular set of conditions the weight of ammonia condensed varies with the piston displacement; but the volumetric efficiency varies with different conditions of head and suction pressures, the quality of the suction gas and the amount of clearance. For example, the real volumetric efficiency may be anywhere from 80 per cent to 50 per cent or less. If the volumetric efficiency (the ratio of the effective stroke to the actual piston stroke) remains the same, it means that the capacity of the compressor (the tonnage) may be increased by speeding up the machine.

The Valves Are Gas-Operated

Ammonia compression is similar to air compression, and the same design, with suitable changes, can be used for either kind of work. However, mechanically operated valves for refrigeration have never been a success, on account of lack of constant conditions as regards both suction and head pressure, and also because of the extra difficulty due to having additional packing in the cylinder where the valve stems pass through to the valve chest. The valves are therefore gas-operated, of one design or another.

High-speed compression must have a special form of valve. The reason for this is easily seen on consider-

ing the problem. It will be remembered that the suction and discharge valves are opened and closed by the action of a difference of pressure. In the case of the suction valve the pressure must be less on the inside than on the outside of the cylinder, and the discharge valve requires a greater pressure on the inside than on the outside of the cylinder in order to open. This difference of pressure increases with the weight of the valve and the speed with which it is opened (the effect of inertia). Again, as the piston speed increases, the size of the valve openings must increase so as to allow the gas to pass in or out with the least practicable effort.

The effect of inertia can be seen by common examples: A boat in the water, or a train of cars, requires some time at full power to attain even moderate speed. A paper may be jerked from under a book, which if moved slowly could have carried the book along. The skater or the "flivver" starting from rest requires full power to attain speed quickly. These examples are all subject to the law that the force required to change the velocity of a body varies as the product of the weight of the body and the change of velocity.

So it is in the case of the valves of ammonia or air compressors. As the speed increases, the force necessary to open or close the valves quickly becomes excessive. The sudden application of the force necessary for the purpose tends to weaken the material also. For these reasons the demand is for a type of valve that is light in weight, a demand that has been met by means of the ribbon, feather or plate valves.

As has been already mentioned, the valves used for ammonia compressors are those which have found success in their application to air compression, both in moderate pressures and the nominal pressures required for blast-furnace compression. One of the most widely known of these is the plate valve.

The plate valve, as its name suggests, usually is made of a thin sheet of steel, is hardened and desirably tempered, and is ground down to a correct flat surface. In the older types, the Borsig and Gutermuth valves, and the more recent design, the Mesta valve, flexibility is obtained by means of perforating the plate (after the principle of a spiral) so that the movement of the plate is easy of accomplishment.

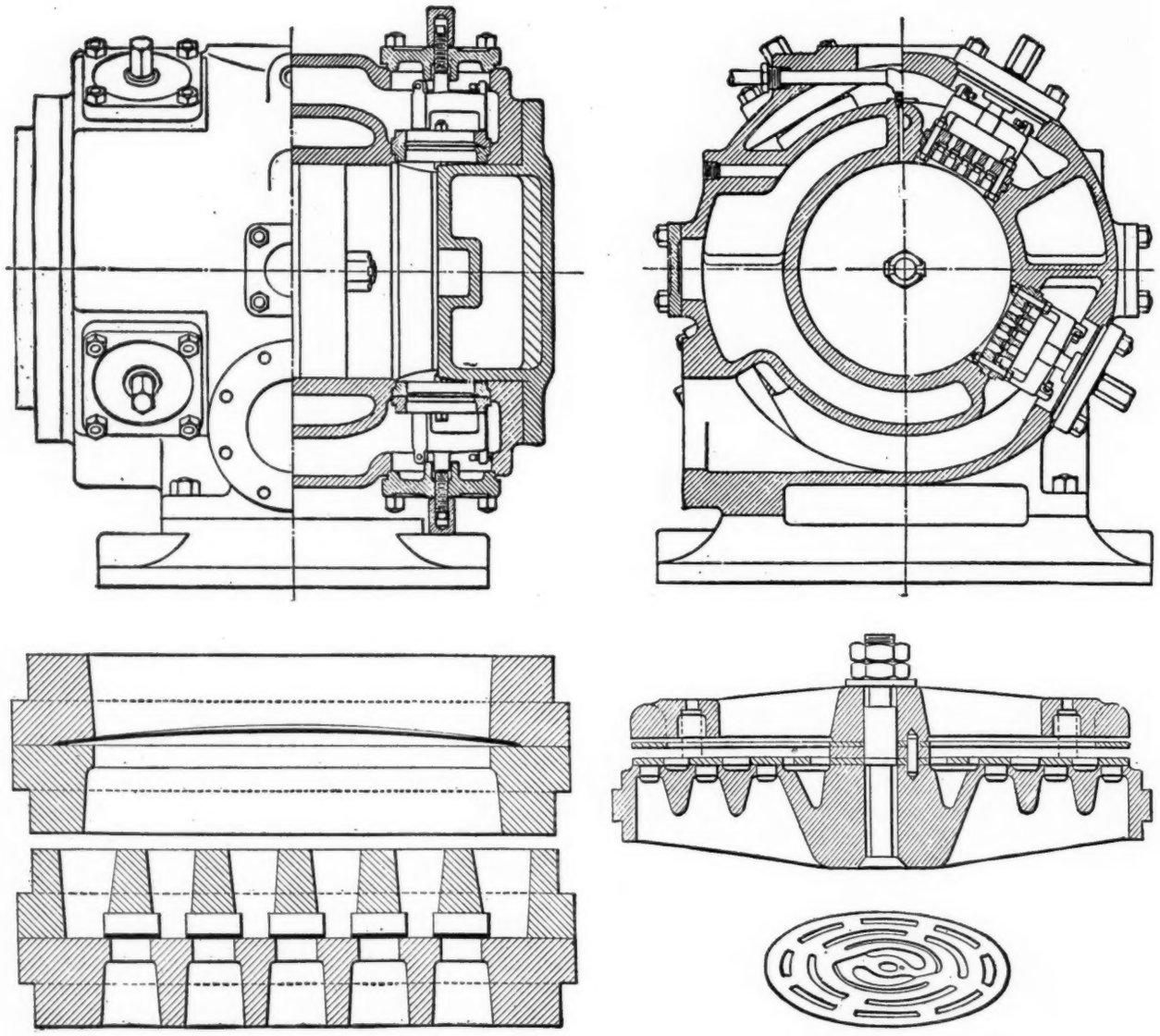


FIG. 1. THE FEATHER VALVE IN SECTION AND AS USED IN THE COMPRESSOR CYLINDER

The main objection to such a design is the fact that the valve is fastened to the seat by means of a machine screw attached to a central part of the spring. This means that the valve lifts off its seat by a bending action similar to the deflection of a cantilever. The operating result is trouble due to occasional breaking on account of repeated stress along the position of

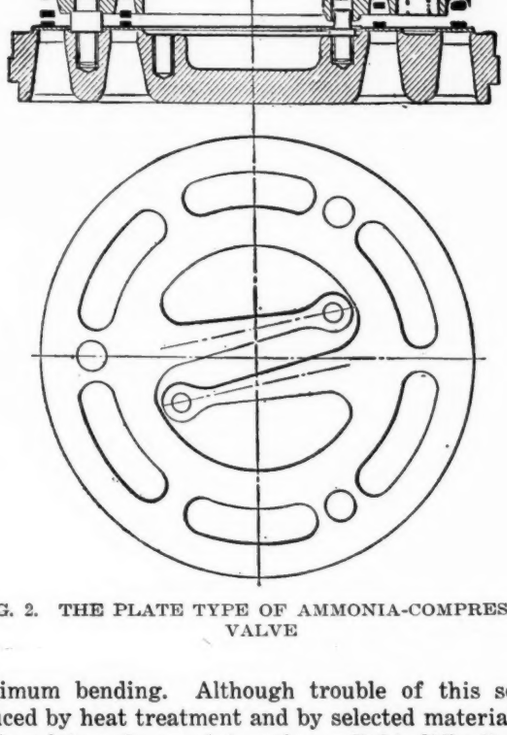


FIG. 2. THE PLATE TYPE OF AMMONIA-COMPRESSOR VALVE

maximum bending. Although trouble of this sort is reduced by heat treatment and by selected material used in the plate valve, and is only a slight difficulty used with air compression, yet for ammonia this type of valve has not been entirely satisfactory. One of the latest of the plate valves is a distinct variation from previous plate valves. It is more like a disc and is flexibly held on its seat by means of a spring. In consequence, there is no call for deflection of the material, and no causing of fracture on that account. The valve lifts off its seat more like a disc valve in the water end of a pump.

Valves of the ribbon or feather type have not come into common use at present for ammonia compression. They have been used in certain special cases, but as yet have not been put on the market in quantity. However, their success warrants further consideration on the part of the engineers.

The Plate Valve in Action

There seems to be a difference of opinion in regard to the relative advantages of the various types of valves. The main objection to the use of the plate valve seems to be in the alleged large resulting unbalanced pressure.

Referring to Fig. 2, it will be seen that the plate valve has to overlap the ports enough to secure a firm seat. This amount may be ⅛ in. all around the port. The result is that some engineers claim that an unbalanced pressure is present, due to the much greater area of valve on the top as compared with the area subject to vapor pressure on the underside of the valve. This difference of area would appear to be from 25 to 35 per cent of the port opening. If such a condition prevailed, the compression would have to be carried to an amount greater than the condenser pressure by a similar amount, as for example, 200 lb. in the case of a 150-lb. condenser pressure. However, it is not clear that such a condition exists, as may be seen by the indicator diagram shown, which is taken from an actual test. Although the writer has seen some indicator diagrams which show that an excess pressure is obtained, the majority of diagrams show little pressure in the cylinders in excess of the head pressure.

In the horizontal compressors using the plate or feather valve, the suction and discharge valves are identical, and frequently of the same size, but the vertical high-speed compressors usually have some different type of valve. For vertical compressors, if the old-type balanced valve in the piston head is used, then the poppet discharge valves (usually a number of small valves) are used. If the plate valve is used, then a modified plate valve or valves using a special dashpot attachment for quiet seating is made use of. The latter valve has been especially successful at high speeds, and 400 r.p.m. has been obtained without undue noise. The valve is not one of extreme thinness, like the early air-compression valves or even the older plate valves, but like the more recent plate valve, it is quite appreciably thick and depends on the cushioning effect of the dashpot to reduce the final velocity of closure.

Quite different from the so-called plate valve, but similar in its action, is the "feather" valve. This is

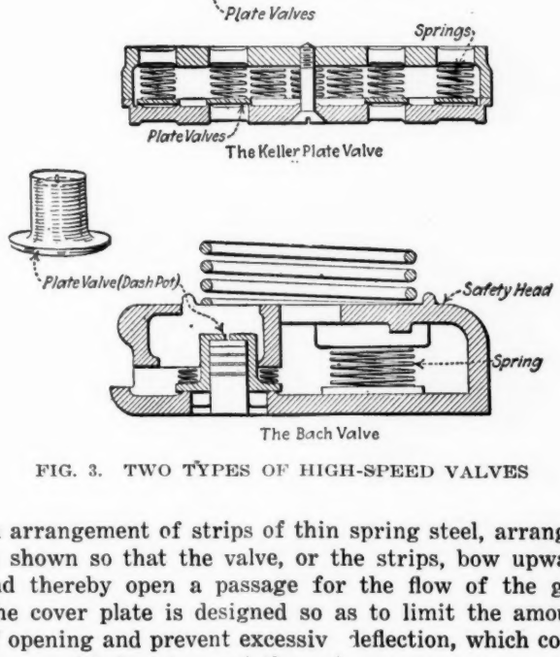


FIG. 3. TWO TYPES OF HIGH-SPEED VALVES

an arrangement of strips of thin spring steel, arranged as shown so that the valve, or the strips, bow upward and thereby open a passage for the flow of the gas. The cover plate is designed so as to limit the amount of opening and prevent excessiv deflection, which could cause quick breakage of the valves.

Clearance

For years we have heard of the dire effects of clearance, and how this should be reduced to a minimum. It is true that clearance will reduce the capacity of the compressor, but the power input (horsepower per ton of refrigeration) need not be increased appreciably. This conclusion may be readily seen from the following: The gas remaining in the clearance volume at the

end of the compression stroke begins to expand on the return stroke and does *work* on the piston during the expansion. Of course there are factors like compressor friction and faulty expansion, but the energy returned on re-expansion is nearly equal to the work of compression. The result is that it is only necessary to increase the piston displacement when the clearance becomes greater than ½ or 1 per cent. The result is a larger engine, or one using higher relative speeds than if small clearances were possible or practical.

On the other hand, high speed requires careful balancing of the reciprocating parts and enough clearance to bring the compressor to rest at the end of the stroke without knocking. Incidentally, the plate form of valve, because of the cages used to secure it accurately in position and to enable it to function properly, requires a clearance of 4 to 7 per cent. The re-expansion under such conditions varies with the pressure range carried in the compressor (the ratio of the head to the suction pressures) and may be from 25 to 50 per cent. As mentioned, this cuts down the tonnage of the machine which is proportional to the weight of gas entering the *suction valve* each stroke, but the cost of a ton of refrigeration is only slightly greater. The tonnage lost by clearance may be made up, however, by increasing the speed.

Of late years there has been a steady increase in the demand for electric-motor-driven ammonia compressors and for direct connection to the motor. It will be remembered that rope and belt drives are often inefficient, varying as they do from 85 to 95 per cent. In the larger compressors it is possible to secure low-speed motors so that direct connection is quite practical. However, the induction motor, because of its small air gap, cannot be used, and recourse has to be made to the synchronous motor, which has a large air gap and the property of operating with unity power factor if desired. Direct connection to oil engines is employed to some extent although it is not general.

At present the main troubles with this type of compressor are those due to faults in the design and lack of care on the part of the operating engineer. The compressor for high-speed conditions requires proper lubrication at all times. The machine work must be accurate, the lubricating system must be adequate and the material annealed or heat-treated. Finally, the compressor must be balanced.

Seattle's two municipal generating stations, with a combined capacity of 50,000 kw., will be increased by the erection of a new plant. Work has already been started on the first unit, which provides for a capacity of 36,000 kw. The power site is on the Skagit River, 100 miles northeast of Seattle, and electrical energy at 160,000 volts will be transmitted to the city.

Carrick Graduated Combustion Control

In burning coal there should be no sudden changes in draft. Movement of the damper position should be gradual, not closed suddenly and then fully opened as in automatic regulation of the open-and-shut type. When there is a bright fire on the grate and the load drops off, the steam pressure increases. The operator or the regulator closes the damper. Immediately the furnace is transformed into a gas producer owing to the extreme change in the air supply, which has been suddenly reduced, and carbon monoxide, CO, results in the flue gases. Then the load increases, the steam pressure drops and the damper is opened. The combustible gas that has been bottled up is released up the stack unburned, an excess of air is admitted, and there is a decided waste. In either case there is a loss in economy, and with this happening many times throughout the day there is a loss of fuel.

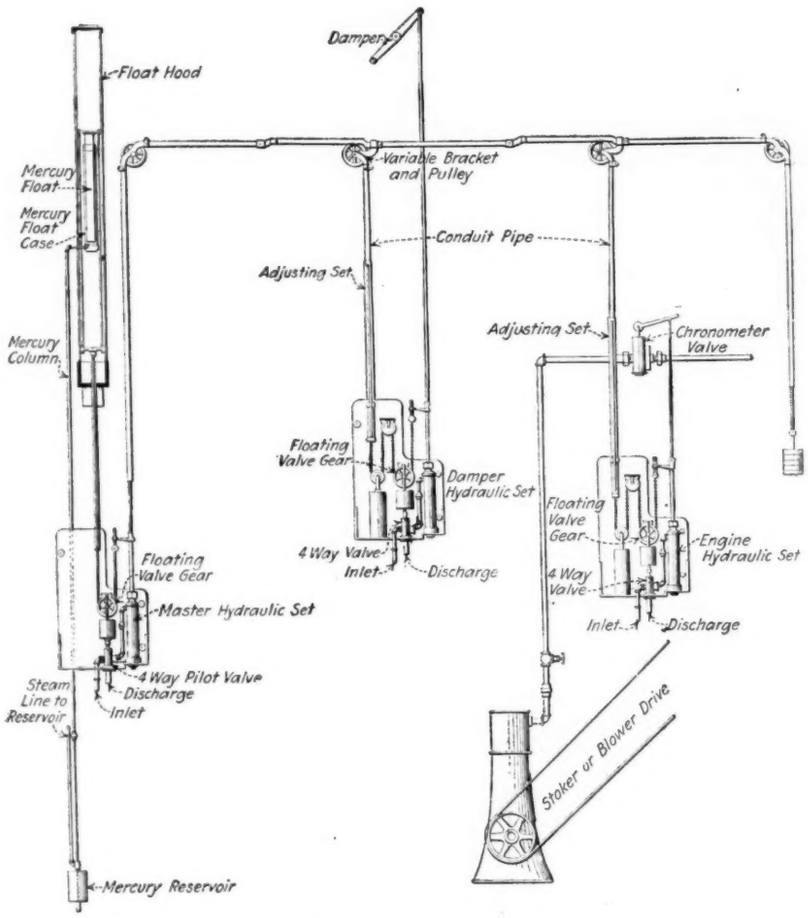


FIG. 1. DIAGRAMMATIC VIEW OF CARRICK SYSTEM OF GRADUATED REGULATION, SHOWING THE MAIN AND AUXILIARY CONTROLS

To meet the requirements, automatic control must function so as to graduate over a predetermined range and to hold the damper in a floating position; that is, hold the damper in a definite position for a definite set of conditions and move it to a new position for a change in any of the variables such as flow of steam, feedwater supply or furnace condition. Stoker speed or coal feed should change an amount corresponding to the change in the damper position, and the coal feed and air supply should be changed simultaneously. Every

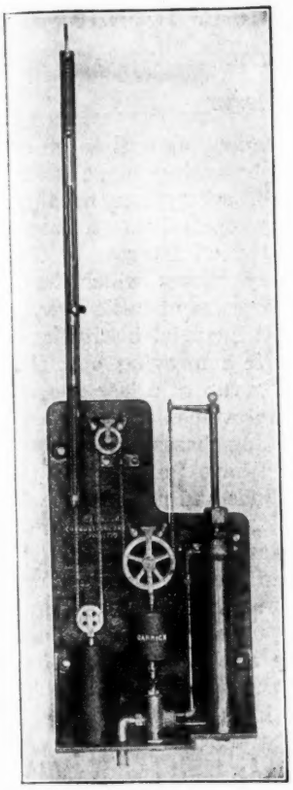


FIG. 2. TYPICAL AUXILIARY CONTROL

damper, stoker engine or motor and forced-draft blower should be controlled individually or so arranged that the relation of the damper or blower to the stoker speed can be readily changed and properly correlated.

A system has been developed recently by the Carrick Engineering Co., of Chicago, to fulfill these requirements. It controls the fuel feed and damper position jointly to correspond with the load, holding the proper relation for efficient results. Owing to a graduated movement it takes and holds any intermediate position that is required for the load carried and moves to a new position if there is a slight change in the load or steam flow, in feed water, or in furnace conditions. This system is designed to maintain the proper relation between the fuel and air supply by individual control of the stoker engines or motors, dampers, forced- or induced-draft fans.

There is a master control with a column of mercury using the steam pressure to actuate the travel of the mercury instead of atmospheric pressure. The column of mercury is two inches high for every pound of steam pressure, so that for every ounce of pressure change there is one-eighth inch of mercury travel.

Referring to Fig. 1, a mercury reservoir shown in the lower left-hand corner is connected by a column with an upper mercury cylinder which contains a free moving float. The latter, by means of a yoke and side rods, has cable connection to the arm of the hydraulic cylinder, shown at the right of the control panel. The chain connection between the floating plunger yoke and the hydraulic piston passes around the floating wheel or valve gear shown at the center of the panel. Just below this wheel is a four-way pilot valve which controls the flow of water from the city pressure either to the top or the bottom of the hydraulic cylinder, depending on whether the steam pressure rises or falls, thus raising or lowering the piston of the valve. It will be noticed that the wheel lever is free to move with a rotating motion as well as up and down, due to a slot provided in the bracket supporting the wheel. Also the hydraulic piston cannot move until the four-way pilot valve is opened.

Assuming an increase in steam pressure, the result is an immediate increase in the mercury level in the upper well or cylinder; the plunger will rise just as soon as the change in steam pressure reaches one ounce, the movement being one-eighth inch. This will raise the wheel, slightly turning it and at the same time raising the entire valve gear and the pilot valve piston, as the hydraulic piston must remain stationary until the foregoing operation takes place. As soon as the piston of the pilot valve is raised, water from the city mains is admitted to the top of the hydraulic cylinder, causing the piston to move downward, lowering that side of the

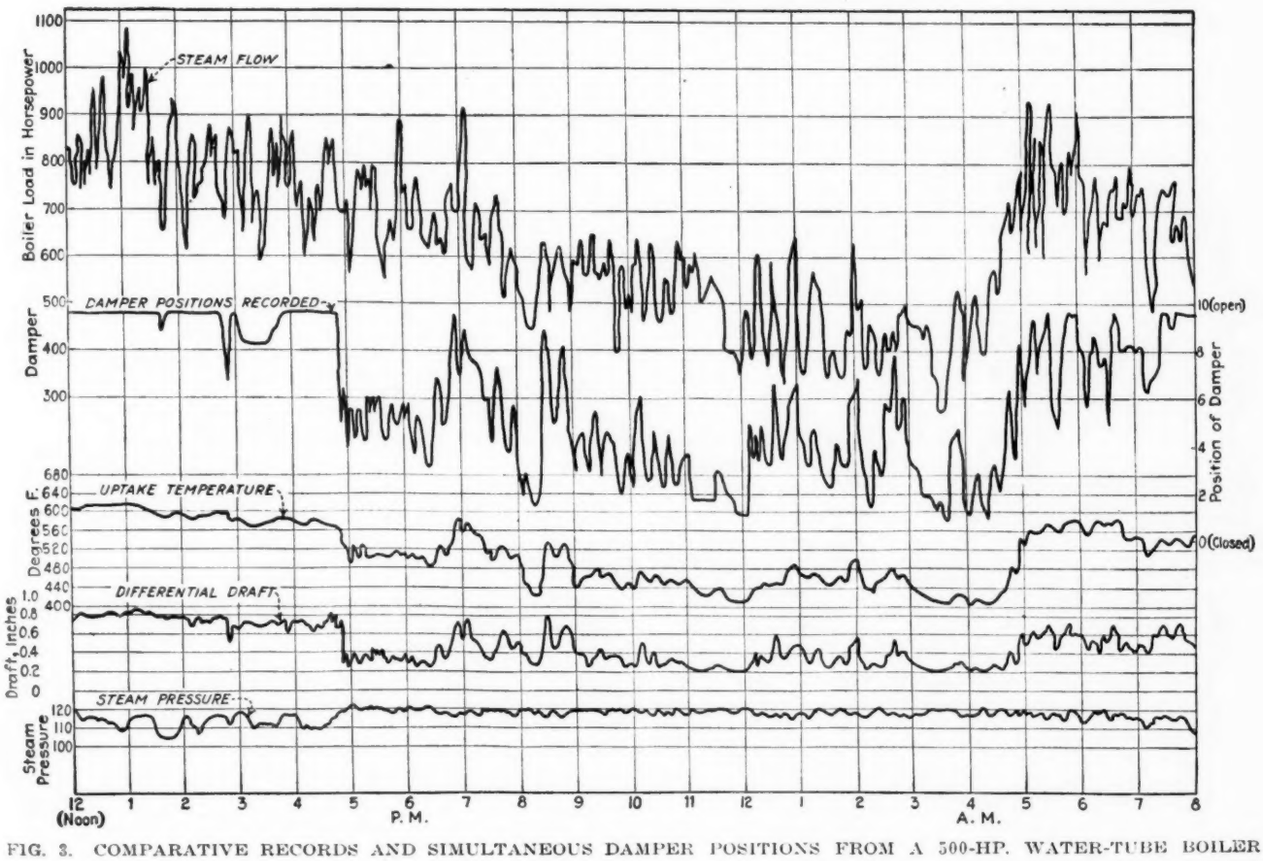


FIG. 3. COMPARATIVE RECORDS AND SIMULTANEOUS DAMPER POSITIONS FROM A 500-HP. WATER-TUBE BOILER

wheel lever and automatically returning the piston of the pilot valve to a neutral or closed position.

When this balance takes place, the master control holds that position until there is another change in pressure. A drop in pressure operates just the reverse. The mercury plunger is lowered; the piston of the pilot valve is lowered; water is admitted to the bottom of the hydraulic cylinder and the hydraulic piston moves upward, pulling the valve piston back to neutral, when there is a balance between the mercury and the hydraulic.

The operation of the master control is transmitted to each individual auxiliary (see Figs. 1 and 2), there being one auxiliary for each individual damper, stoker engine or motor, forced-draft fan, etc. These auxiliaries are all connected to the master by a rope or chain, and the connections are run through conduit pipes with special conduit pulleys so constructed as to make it possible to turn an angle of any degree. Where an auxiliary is installed, a special insert is provided so that a break can be made in the main conduit line and a branch cable taken off for the auxiliary. As the master moves, so moves each auxiliary, thus giving individual control to each particular part of the equipment that enters the combustion process.

One of the features of the graduated control is the adjusting set that is used on each auxiliary. By means of this adjuster one boiler can be set ahead or back of the others, and the load can be carried on a certain number of boilers up to a predetermined rating, and the other boilers can then be cut into service automatically and at any rate desired.

Where engines or turbines are used for driving stokers or fans, a special chronometer valve is employed, which is designed to supply steam under graduated control. Roughly, the valve is of the vertical piston type with a rectangular opening through the piston and a corresponding rectangular port in the body of the valve. The length of stroke of the piston is determined by the auxiliary control so that the flow of steam to the stoker or fan engine will correspond with the requirements. There is also provision for micrometer hand adjustment to turn the piston of the valve and thus vary the width of the port to suit the requirements of any particular machine.

An illustration of the operation of the control system may be had from Fig. 3, which presents comparative records taken on a 500-hp. water-tube boiler for twenty consecutive hours. The load chart is at the top and the damper position just below. Other simultaneous readings that have been plotted are uptake temperature, differential draft and steam pressure.

Notice that at 7 p.m. for the load of 900 hp. the damper is nearly wide open. From then until about 8:30 the load diminishes and the damper position changes to correspond. Also between four and five o'clock the next morning the load gradually increases, and again the damper changes its position, gradually opening up little by little, until when the peak is reached, the damper is in the right position to supply the proper draft and produce the rate of combustion necessary to carry the load efficiently. The speed of the stoker has been increased in the same proportion so that the efficient relation between the draft and fuel bed established for the grade of coal burned, has been maintained automatically and with much closer precision than is possible with hand regulation.

Financing Improvements for the Power Plant

BY PAUL R. DUFFEY

Many security and trust companies, as well as other financial institutions, are daily investing large sums of money in the many classes of enterprises needing a working capital to make them operative. A large majority of such enterprises net good interest to the investors. Some fail from many causes which tend toward failure even though the investment looked good.

The thought appeals to me that financial institutions would do well to invest money in improving obsolete power plants which are now being run at a loss due to lack of capital for necessary improvements, and place the plants on up-to-date operating bases. My idea is for the financial institutions to have their engineers go over the plants applying for funds and make detailed reports of present operation, etc., and place the proposed requirements for proper consideration.

Citing for an example a steam generating plant of around two thousand boiler-horsepower, the boilers being hand-fired and the plant under test shows an over-all boiler furnace and grate efficiency of from 54 to 58 per cent. The total cost per year for operating this plant, present and proposed is as follows:

	Present	Proposed
Labor	$12,467.84	$6,679.16
Fuel	49,131.20	39,304.99
Handling fuel	1,200.00	600.00
Water	1,686.15	1,686.15
Maintenance	2,706.62	2,706.62
Supplies	300.00	300.00
Total per year	$67,491.81	$51,276.92

Present operation, $67,491.81; proposed operation, $51,276.92; saving per year, $16,214.89.

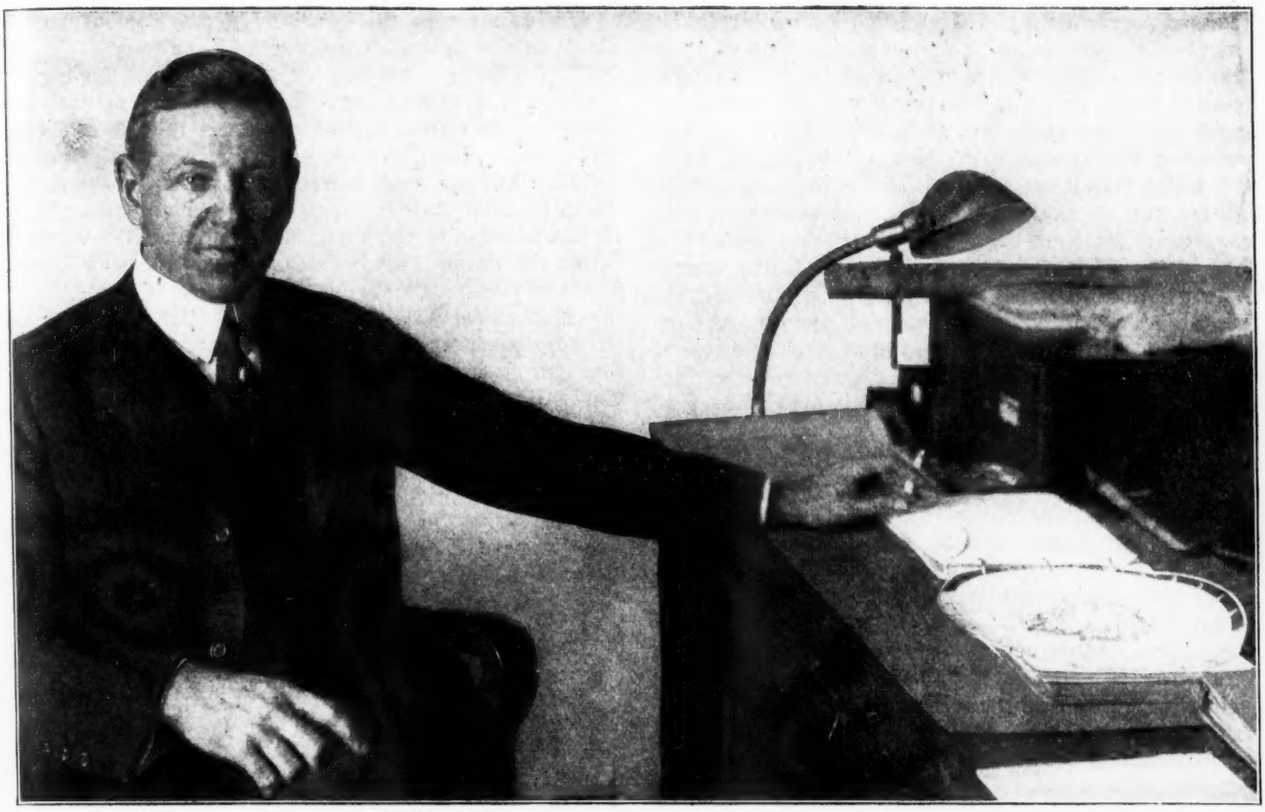
Overhead charges have been disregarded, as they would remain about same as at present.

In order to bring about this saving, which would build the over-all efficiency of the plant up to 75 per cent, a capital investment of $33,785 is required. It is proposed that the financiers invest this amount and have the plant operated at the new figures of $51,276.92 per year and that they receive from the industry for which they have made the improvements the sum of $67,491.81 per year, the present operating costs, for a period of time sufficient to net 10 per cent on the investment of $33,785 and that as soon is the latter amount plus 10 per cent interest has been earned, the plant should be turned over to the owners in full. Thus the owner would have in, say, three years the new up-to-date plant fully paid for out of the savings effected.

Any additional cost of labor, fuel and supplies during this period is to be borne by the company on a basis of the old plant figures or present costs. In other words, this is simply purchasing a plant on the installment plan, the same as other more or less familiar commodities.

During the past few years the number of gasoline-consuming engines and vehicles, including pleasure cars, trucks, motor boats, tractors and farm engines, has increased to such an extent that there is danger in the near future of an acute shortage of this most important fuel. In the United States alone there are 4,000,000 automobiles, 250,000 trucks, 500,000 motor boats, 75,000 farm engines, making a total of 5,575,000 gasoline-consuming engines.

R. J. S. Pigott—Power-Plant Economist



Present fuel conditions, involving both shortage and high price, indicate that power users, and especially industrials, must adopt a course of improvement in plant design and equipment, as has been the case in Europe. Formerly, power cost in many industrial establishments was so small a percentage of manufacturing cost that it received but the scantest attention and economy was but little observed. It will become a vital question whether the isolated plant or the central station is the ultimate solution. In all probability some development such as W. S. Murray's super-power system will occur, but can supplant the isolated plant in general only if some of the inherent advantages of the latter are also developed in the central-station system; namely, the large use of low-pressure steam for heating and processing and the use of refrigerating plants and similar service for off-peak load. In short, the percentage of thermal recovery from the fuel must be increased by the development of all the services now given by an isolated plant to its factory, before the central idea can give a real fuel economy, due to its inherent advantages of size diversity factor and higher-grade technical supervision.

DURING the year 1908 it became evident that in a very short time additional power would be required for the New York subway. The Fifty-ninth Street power plant of the Interborough Rapid Transit Co., in which the subway power was generated, was equipped with nine compound engines, having a maximum capacity of 7,500 kw. each. A careful study of the situation resulted in a decision to install a 7,500-kw. low-pressure turbine in connection with each engine. The importance of the station and the success of the new arrangement attracted nation-wide attention to the work and to the man who directed it. The superintendent of motive power was H. G. Stott, but details of design and construction were largely the work of R. J. S. Pigott, chief draftsman, motive power department, and later assistant engineer in charge of construction.

Mr. Pigott, who was born in England in 1886, was graduated from Columbia University in 1906 with the degree of mechanical engineer. In that year he became an assistant engineer in the motive power department of the Interborough Rapid Transit Co. and the next year was made chief draftsman in the same department. It was while he was in this position that the plans for the low-pressure turbine installation were taken up, and he had an opportunity to show his ability in the handling of real power problems. This job involved not only the erection of the new turbine units, but a rearrangement of the piping and condensing equipment to comply with the changed conditions.

Since the whole plan was to a large extent an experiment, one turbine was installed and numerous tests made on the combined unit to determine how nearly the entire plan might be expected to come up to the predicted performance. This test work alone presented a number of new problems for solution. One of these involved the determination of the quality of the low-pressure steam going to the turbines. As a result of considerable experimental work the conclusion was reached that the ordinary perforated-pipe sampling tube, which was found to give satisfactory results with high-pressure steam, really acted as a separator with low-pressure, so that the sample taken through such a tube was very far from being a representative sample of the steam in the pipe. A sampling tube was designed with a single orifice pointed against the direction of steam flow and proportioned so that the velocity of steam in the tube should be very nearly equal to that flowing in the pipe. It is interesting to note that Professor Rateau, the famous French engineer, has taken occasion to confirm Mr. Pigott's findings in connection with the investigation of the quality of low-pressure steam.

In 1911 Mr. Pigott became superintendent of construction for the New England Engineering Co., and in 1912 professor of steam engineering at Columbia University. In 1913 he returned to the Interborough Rapid Transit Co. as construction engineer. This position he held until 1915, laying out and installing three 30,000-kw. turbines with all other plant changes. During his two periods of service with this company Mr. Pigott was responsible for the design and installation of a number of appliances and procesess which resulted in improved power-production economy. Among these was the design, some years ago, of a traveling screen for use at the Fifty-Ninth Street station in the circulating-water inlet. This is believed to be the first use of this type of screen in power-station work. At the Seventy-Fourth Street station of this company Mr. Pigott did considerable work in connection with the use of furnace-wall ventilation to prevent burning out linings in boiler furnaces.

In 1917 Mr. Pigott went to Bridgeport, Conn., as consulting engineer and power superintendent for the Remington Arms Co. At that period the enormous expansion of the Remington works called for by the war-time emergency required a complete design for the new plant. Mr. Pigott undertook the design, construction and operation of all power and plant service systems, and in doing so contributed an important bit to the country's war-time needs. During this period he also found time for work in the reorganization of the production departments of the Bridgeport Brass Co. and for the development of design and construction data for the Sanford-Riley Stoker Co.

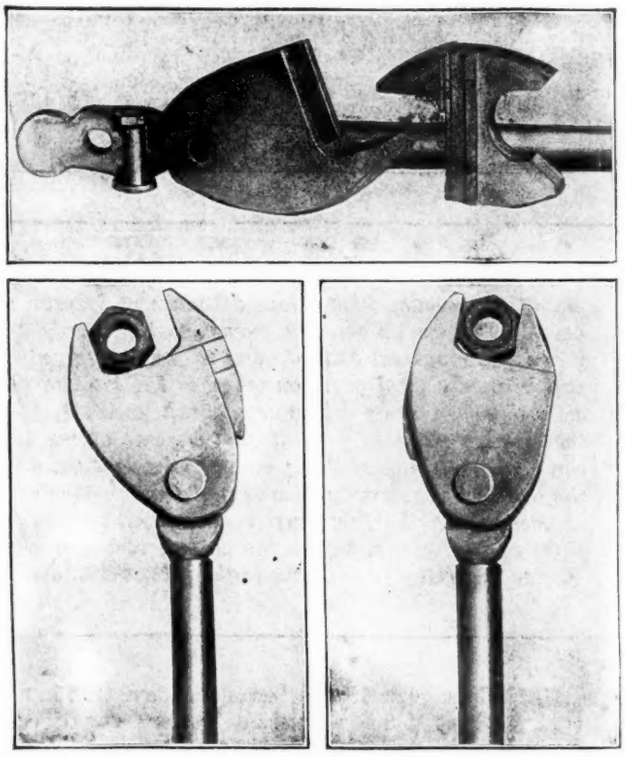
While the requirements of his work have at times turned Mr. Pigott's attention to other fields, his principal interest has always been in the field of economical power production. In a recent interview he describes himself as a power-plant man and nothing else, but his success in the solution of problems along broader lines shows him to have ability in other directions.

Since 1919 Mr. Pigott has been industrial engineer for the Bridgeport Chamber of Commerce. In discussing recently the questions of fuel economy, he stated that one of the arguments that prevail against the small isolated plants is the inability of such a plant to support the necessary expert supervision to produce power to the best advantage. In answer to a question he said that his work for the Bridgeport Chamber of Commerce consisted to a large extent in furnishing advice with regard to improvement in fuel economy to the various power plants in and around Bridgeport.

Mr. Pigott is a member of the American Society of Mechanical Engineers, of the American Institute of Electrical Engineers and of the Society of Automotive Engineers. His membership in these professional societies has not been merely passive, as is shown by the large amount of committee work which he has done. He is chairman of the Fluid Meters Committee, a member of the Power Test Code Committee, formerly chairman and present member of the Research Committee of the Mechanical Engineers. During the war he was advisory engineer for the Connecticut Fuel Administration and chairman of the Bridgeport Committee of the Naval Consulting Board, and also engineer for the Bridgeport Smoke Abatement Commission.

Adjustable Ratchet Wrench

A self-adjusting ratchet wrench that will take the place of several sizes of open-end wrenches, especially designed for heavy-duty work and a time saver, due to the self-adjusting feature and the ratchet action, is



PARTS AND APPLICATION OF RATCHET WRENCH

being brought out by the Allan-Diffenbaugh Wrench and Tool Co., 6250 South Halsted St., Chicago, Ill. The parts are interchangeable. Referring to the illustration, the upper view shows the parts, the lower left view shows the jaws wide open, and the lower right view shows the jaws closed on a nut. To operate, simply place the jaws over the nut, and pull on the handle; the harder the pull the harder the jaws will grip. To let go of the nut, release on the handle, which will loosen the grip of the jaws and permit a ratchet action on the nut.

EDITORIALS

Distance of Boiler Above Grate

SOME weeks ago a reader inquired how high a seventy-two-inch horizontal return-tubular boiler should be set above the grate. The editor answered thirty to sixty inches for ordinary fuel and hand-firing. A correspondent thought it should be from four to five feet, and the question has been the subject of considerable discussion.

It is not so many years ago that the editor's figures would have been considered excessive. The average height above the grate of the six seventy-two-inch boilers treated in "Barrus' Boiler Tests," published a quarter of a century ago, is less than twenty-seven inches. The catalogs of several large builders of boilers of this type give from twenty-seven to thirty-four inches. Up to the latter part of 1914 the Hartford Steam Boiler Inspection and Insurance Company recommended twenty-eight inches as the minimum distance for a seventy-two-inch boiler, but has recently revised its practice to thirty-six inches for anthracite and forty for bituminous coal. Osborn Monnett, who attained considerable reputation as a furnace and combustion expert while at the head of the smoke-prevention department of the City of Chicago, says not over thirty inches, even for Illinois coal, and Mr. Misostow, his successor, is for a lower value still. Several of the boilers that were set forty-eight inches above the grate in the days when Chicago was experimenting with smoke prevention have been lowered with improved results.

Is it not, after all, a question of rate of combustion? Is not a certain minimum volume necessary for the most effective combustion of a given weight of carbon and hydrogen in a given time, and is not increased height, and thus greater volume, necessary for a furnace in which high rates of combustion are used, although the lower settings in common use may be sufficient for more moderate rates of firing? The average rate in the Barrus tests was less than twelve and one-half pounds per square foot of grate per hour, and the average capacity only slightly above seventy-five per cent of the rated.

Much stress is put upon furnace volume where powdered coal is used. In oil-fired Babcock & Wilcox boilers sixty to eighty cubic feet per burner with a maximum capacity of three hundred and fifty pounds per hour is supplied. This is from 0.17 to 0.23 of a cubic foot per pound. The same builders allow ten cubic feet of furnace volume per square foot of active grate area for multiple-retort underfeed stokers. This would be 0.25 of a cubic foot per pound for a firing rate of forty pounds of combustible per hour.

A high setting means greater first cost, more chance for radiation and air infiltration and a less immediate and perhaps less complete use of the radiant heat. The contention of the advocate of the low setting that the boiler gets more heat from radiation is met by the assertion that the heat, once it is generated in the furnace, cannot get away and will be absorbed in the tubes if it is not in the fire sheets. It may be but the low-setting man prefers to take advantage of the direct penetrating effect of the radiant heat rather than to depend upon getting the heat into the boiler by convection. If the boiler is near enough to the fire, a large percentage of the steam will be made on the fire sheets, and this largely by radiant heat which can penetrate the layer of cooled gas next to the shell. In this way the heat is certainly absorbed; if convection is depended upon, it may if the tubes are clean enough inside and out. The essential is that the boiler must be far enough from the fire to give the gases room and time to burn completely before they are chilled by contact with the comparatively cool shell, and this distance should obviously be greater when forty or more pounds of coal are burned per square foot of grate per hour than when the rate of combustion is fifteen pounds or less.

Motors for Driving Ventilating Fans

INTIMATELY connected with every power plant is a ventilating problem of some kind or another. This may take the form of supplying fresh air to a large building or to a small room, or to industrial processes. However, in a very large percentage of cases motor-driven fans are employed. The ordinary ventilating fan is a comparatively simple piece of equipment from the structural standpoint, but if satisfactory operation is to be obtained not only must the fan be driven by a motor that is suited to the application, but a controller that will give proper control of the motor must be employed. In this issue, on page 759, W. H. Easton discusses "Motors for Driving Ventilating Fans" and gives a very clear and practical analysis of this phase of motor application.

Too much cannot be said regarding the location of the motor. On account of the nature of the work it is frequently necessary to place the machine in an out-of-the-way place, although this does not necessarily mean that it must be in an inaccessible location. If it is so placed that it is difficult to care for, it will probably be neglected, with the result that the service will be impaired and the maintenance charge will be high. The placing of a motor in an inconvenient place because nothing else will go there is false economy generally and results in dissatisfaction. The question of motor efficiency in regard to fan drives is an important one and involves two factors—the efficiency of the motor when operating under normal speed and the efficiency at other than normal speeds. Although the latter is to some extent influenced by inherent characteristics of the motor, the largest factor is the method of control, especially with direct-current motors. If, on account of its lower cost, a motor is selected, where adjustable speed is required, that will, when operating at normal speed, drive the fan at the maximum required speed,

and it is necessary to operate for considerable periods at reduced speed, low efficiency will result.

The cost of the power lost in the armature external resistance will usually pay a handsome dividend on the additional capital required to install a motor and controller that, at normal speed, will drive the fan at the lowest speed required and beyond this point adjust the speed by field control. With alternating-current motors the problem becomes more involved. The simple squirrel-cage type of motor, although being the most reliable in its operation and of high efficiency at normal speed, is strictly a one-speed machine, therefore, where adjustable speed is required, this type must give way to the wound-rotor machine, which, when operating below its normal speed, has an efficiency characteristic similar to the shunt direct-current motor with armature control. There have been developed special types of induction motors for speed regulation, but they have not yet come into general use for fan drives. However, the increased cost of power no doubt will cause greater attention to be given to these more complicated equipments as a means of obtaining a higher efficiency in alternating-current motors at different speeds than can be reached with the wound-rotor type.

Storage of Winter Coal Supply in Summer Is Still Important

EFFORTS to stabilize the coal industry by buying and storing coal for next winter's use were given a setback in some cases by the outlaw railroad strike. For a time at least the roads could move neither full nor empty cars, with the result that even present demands could not be met and in many cases both the mines and the coal-using industries were compelled to shut down. This should not discourage those who are planning to do their share toward the general good and to help themselves by anticipating their extra coal requirements. The strike situation will soon be cleared up, and coal ordered can be mined and delivered.

Meanwhile, the bill allowing differential freight rates for coal in the spring and summer months is in the Senate committee. In addition, another bill providing for a flat discount instead of the percentage discount, as called for in the first bill, has been introduced in the Senate. This discount varies from month to month during the low-demand period of the year. Whatever may be the fate of these bills, they will probably be side-tracked until the present political activities of the various Senators have somewhat relaxed. It seems, therefore, that little help may be expected from this source during the present season.

On the other hand, the general freight-rate increase seems bound to come, and any coal user who can get his coal hauled before these new rates go into effect can make a considerable saving. At the present time the probable increase is estimated at about twenty-five per cent.

The important point, however, is to stabilize the coal industry, not just to help the coal industry but to help everybody. Fuel prices will continue to be high, and seasonal coal famines will cause trouble as long as the present practice of seasonal buying persists. The winter peak must be taken off both the mines and the railroads before the coal market can be put on a sound economic basis. And it is important to every individual in the country that it be put on such a basis.

The Technical Man as an Inventor

DURING the war a great deal was written about the wonderful things to be accomplished by American inventive genius. It was confidently predicted that when the American people put their minds seriously to the task, the German submarine problem would be solved overnight. Without denying any credit for the really wonderful work accomplished, it is worth while to consider the enormous amount of effort and money that was spent by individuals in developing ideas that had no possible chance of success.

While the war emergency stimulated the efforts of the would-be inventor, such work is going on continually. It is a pitiful fact, however, that many men with intelligence and pluck are working hard at impossible tasks. Having no conception of the limits imposed by natural laws and no knowledge of the mistakes and failures of others, they go blindly on, reaching for a star.

It is in this field of invention and research that the value of a technical or scientific training becomes most apparent. The technical engineer or scientist is trained to determine first a standard of perfection, or the point beyond which it is impossible to go, and then bend all his efforts to approach that standard. He also has, or knows where he can obtain, a knowledge of the successes and failures of the workers who have preceded him. The non-technical man too often accomplishes nothing simply because he wastes his time striving for the unattainable.

We do not claim that the scientific man never makes mistakes. He is not perfect. Neither is it true that the non-technical man always fails. Real progress can be made only by long, patient work, and results usually come only after many failures. However, a knowledge of what has been accomplished and what is contrary to natural laws gives the technical inventor a big advantage.

On April 21 Justice Bailey of the Supreme Court of the District of Columbia issued a temporary injunction restraining the Federal Coal Commission from requiring the Maynard Coal Company to file reports of its costs and other data. While this injunction applies technically only to the case of this particular company, it seems probable that similar injunctions would be granted to other operators in the same position as the Maynard Company. The Federal Coal Commission has been established and its expenses provided for out of the public funds, but the legislation under which it acts is so poorly drawn that the first review in the courts removes its power to collect data. Somebody gets a good job, some members of Congress "point with pride" to an achievement for the benefit of the "people," the "people" pay the bill and there it ends.

The resignation of Dr. Van H. Manning, Director of the Bureau of Mines, is another instance of how inadequate compensation to scientific and technical men in the Federal service is creating a deplorable state of disorganization in the Government departments. In his letter Dr. Manning says: "I feel very deeply that there ought to be more adequate compensation for the scientific and technical men in the Government service so that none of them may be compelled to accept positions on the outside."

CORRESPONDENCE

Eliminating the Smoke Nuisance

I have experimented for many years on various devices to overcome the smoke nuisance. The wet process of water sprays, being the popular one, underwent a thorough trial, and eventually was discarded, and a new theory tried out to determine a velocity of the gases which would be lowered to a point where the force of gravity would overcome the velocity. After much experimentation it was found that should the velocity of the gases be curtailed to a certain given speed, allowing a time element at that speed to take place, all heavier than air particles would be precipitated.

An expansion chamber was designed and built, with these features in view to meet existing conditions, on the stack of 2,500-hp. boiler plant, using buckwheat anthracite for fuel, with forced draft. Properly designed hoppers were arranged to remove the residue of dust and cinder that was collected and under which a track for industrial cars could be used.

The results obtained were beyond expectation, and without reduction of draft in the stack, the residue recovered amounting to 3.77 per cent of the coal used and making a clearance of the flue gases of from 90 to 100 per cent. The original analysis of the coal was 72 per cent fixed carbon, whereas the residue analyzed from 55 to 70 per cent fixed carbon. Evaporative tests run, using this residue for a fuel, adding 20 per cent by weight of bituminous coal to supply volatile carbon, under a separate boiler, gave, at the boiler rating, 8 lb. of water evaporated per pound of the mixture, this result obtained being a little less efficient than the original coal.

This installation of expansion chamber or collector was experimental and naturally more expensive than it would otherwise have been, and even allowing for all discrepancies and expenses conservatively estimated, it produced a dividend on the investment of 29 per cent. This method is just as effective on bituminous as on anthracite fuel and will eliminate the heavier than air particles.

As this device, which is covered by U. S. patent, has no moving parts, and operating costs are nil, the first cost is the only one to be considered. Being beyond the experimental stage, and demonstrated to be perfectly practical, and as it can be adapted to any existing plant, it would seem that from an economical and hygienic standpoint it is no longer necessary to be subjected to the evils of a polluted atmosphere from this cause. THOMAS GRIEVE.

Perth Amboy, N. J.

Reboring Corliss Valve Chambers

With reference to Mr. Young's communication in the issue of April 6 and Mr. Murphy's in the issue of Feb. 17, I offer the following suggestions:

The practice of reboring valve chambers on Corliss cylinders is quite usual and is found necessary when the steam and exhaust valves become worn, thus resulting in considerable leakage of steam and causing the valves to rattle, owing to the excessive amount of play in the valve chambers. This excessive wear is often caused by wet steam, due to foaming or priming of the boilers, which makes it difficult to lubricate the valves properly. This condition can often be improved by installing a good separator in the steam line.

In setting a boring bar for boring the valve chambers, it should be set central to the bore on the valve-gear side of the cylinder. The fit of the front or valve-gear bonnet in the valve chamber should not be disturbed and both the steam and exhaust ports should not be rebored clear through, but the original bore should be left intact for the depth of the boss on the front bonnet. The fit of the back bonnets is not so important, and it really is no detriment if the boss on the back bonnets does not fit the valve chamber exactly. With the front bonnets it is different, as they have to withstand the working strain of the valve gear, and for this reason it is important that the exact fit of the bonnet in the valve chamber be maintained.

It is important in reboring the valve ports that the lowest spots in the wearing surfaces should be eliminated, so as to insure a round chamber, and that the wearing surfaces shall be steam-tight. To accomplish this, it is sometimes necessary to take as many as three cuts. The boring bar should be of the self-feeding type, operated preferably by an electric motor, but of course the motive power can be a steam engine or hand power. With the electric motor the motion of the cutting tool will be much more uniform and steady. After reboring the valve ports, new steam and exhaust valves should be fitted, which would in reality make them as good as they were originally. After starting up the engine with new valves and rebored valve ports, it is important that the lubrication be not stinted until the valves are worked down to a good bearing, and it might be necessary to ease off the hard spots once or twice, so as to facilitate getting the valves to bear evenly over the whole surface. There should be no trouble experienced in work of this character if performed in a careful, workmanlike manner, and there should be a marked improvement in the steam consumption of the engine. JOHN J. MUIR.

Seattle, Wash.

Economical Combustion of Fuel Oil

Although much more has been written on the economical combustion of coal than of oil, there is a very great chance for loss from excess air with the latter fuel. Smokeless combustion of oil can be maintained with very little excess air, but beyond this point there is no outward indication of excess even with three or four times the necessary amount. It is quite necessary, then, to take considerable care to regulate the air control, especially with changing loads on an oil-burning boiler. A CO_2 recorder or other form of flue-gas analysis is necessary for intelligent control of this air supply.

In the plant with which the writer is connected CO_2 recorders are used and are frequently checked with an Orsat apparatus. The highest CO_2 reading we have obtained is 14.9 per cent, and it is interesting to determine how nearly perfect this figure is.

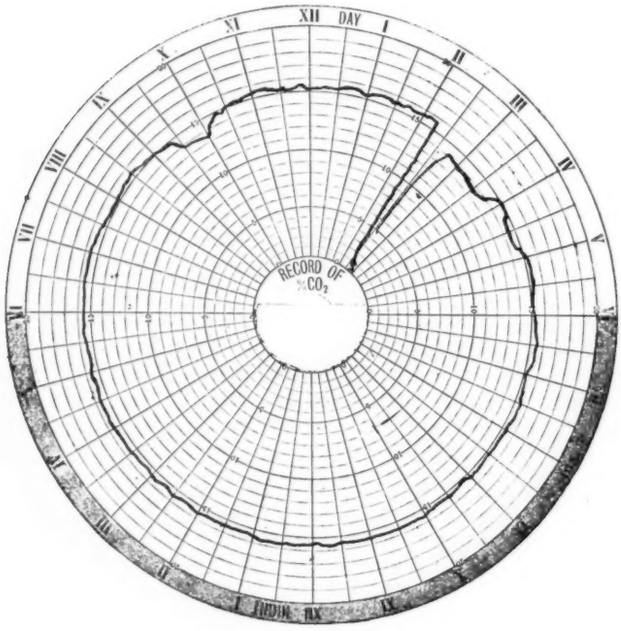


FIG. 1. CO_2 RECORDER CHART FROM BOILER UNDER CONSTANT LOAD

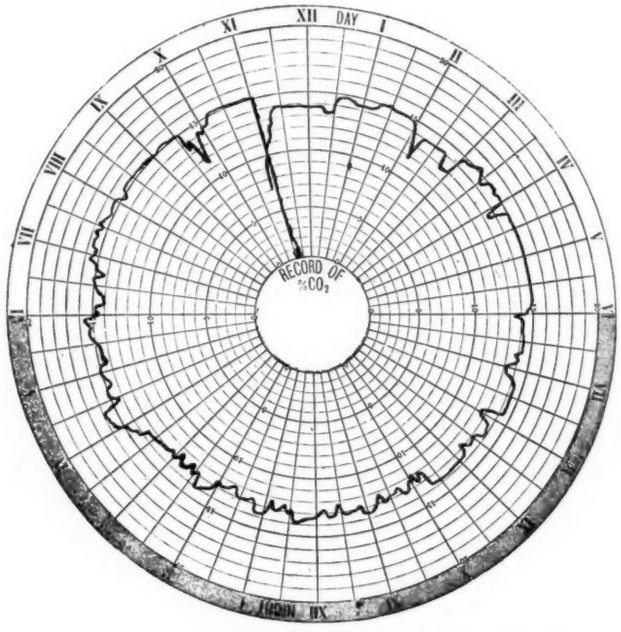


FIG. 2. CO_2 RECORDER CHART FROM BOILER CONTROLLED BY GOVERNOR

Not having an analysis of the oil, an estimate was made from the flue-gas analysis. A typical case showed CO_2, 14.5 per cent; O, 1.5 per cent, and N, 84.0 per cent. Assuming 100 cu.ft. of flue gas, the 1.5 cu.ft. of oxygen will bring with it $\frac{1.5 \times 79.1}{20.9} = 5.68$ cu.ft. of nitrogen. This is the nitrogen in the excess air. Therefore there were 84.0 — 5.68 = 78.32 cu.ft. of nitrogen in the air actually used for combustion. The total oxygen used for combustion is then $\frac{78.32 \times 20.9}{79.1} = 20.69$ cu.ft. The volume of CO_2 is equal to the volume of oxygen required for its formation, so that the oxygen combined with carbon is 14.5 cu.ft., leaving 20.69 — 14.5 or 6.19 cu.ft. to combine with the hydrogen in the fuel.

Since for equal weights hydrogen requires three times as much oxygen as does carbon, the ratio of weights of carbon to hydrogen in the fuel is as 14.5 is to $\frac{6.19}{3}$ or as 7.023 to 1. The percentage of hydrogen in the oil therefore is $\frac{100}{1 + 7.023} = 12.46$ and of carbon 100 — 12.46 = 87.54 per cent. Other flue-gas analyses give practically the same results for this fuel. Of course this does not take into account any oxygen, sulphur or other ingredients in the oil, but the approximation is good enough for the present purpose.

Since the volume of air required to burn one pound of carbon is 152 cu ft. (at 60 deg. F. and atmospheric pressure), the air required for one pound of the above fuel oil will be $(0.8754 + 3 \times 0.1246) \times 152 = 189.9$ cubic feet.

This will contain 189.9 × 0.701, or 150.2 cu.ft. of nitrogen. The carbon alone will require 0.8754 × 152, or 133.1 cu.ft. air, which will contain 133.1 × 0.209 or 27.8 cu.ft. of oxygen. Since the volume of CO_2 is equal to the volume of its oxygen, the flue gas will be made up of 150.2 cu.ft. of nitrogen and 27.8 cu.ft. of CO_2 if no excess air is supplied. The percentage of CO_2, then, will be $\frac{27.8}{150.2 + 27.8} \times 100 = 15.6$. The H_2O formed by burning the hydrogen, of course, being condensed, does not appear in the flue-gas analysis.

This value, 15.6 per cent CO_2, is the theoretical maximum for perfect combustion for this particular oil. The accompanying charts from CO_2 recorders show how closely this condition was obtained. Fig. 1 is from a 286-hp. B. & W. boiler operating under a constant load of 165 per cent of rating, and Fig. 2 is from a 160-hp. return-tubular boiler under variable load with the oil pressure controlled by a governor. Checking the charts with an Orsat apparatus showed that the recorders read a fraction of one per cent high. The variations in performance on the chart in Fig. 2 show the necessity for careful air regulation with changes in load.

The furnaces of these boilers were designed by the writer. They have unusually large combustion chambers with provision for a long flow of gases during combustion. Other furnaces in the same plant cannot be made to give such good CO_2 records.

I believe that proper fuel-oil combustion requires some radical changes in furnace design from present

general practice. The combustion space should increase in volume in the direction of flow of the gases. There should be sufficient volume for combustion to be completed before the gases strike the heating surfaces or bridge-wall, when using the minimum of excess air for complete combustion. Air inlets should be arranged so that all the air may mix with the oil vapor in the early stages of combustion.

In a properly constructed furnace, when correctly operated, the lower part of the combustion chamber will be filled with burning gases, showing a bright yellow and no flame reaching the heating surface. If too much air is used, the flame will shorten and become brighter. There will also be an increase of the roar in the furnace and a perceptible rise in the flue gas temperature.

Ocean Falls, B. C. C. H. LEICESTER.

Changed Stoker Lineshaft

Where a lineshaft is used for driving chain-grate stokers, as is usually the case where one or more batteries of boilers are equipped with stokers, its location becomes a problem, especially where the design includes a siftings hopper for discharging into a chain conveyor.

If the shaft is placed above the stoker, it is in the way of gate adjustments, fuel feeding, boiler washing, etc., and must be protected by safety screens to avoid accident; if placed in the conveyor tunnel in a position to clear the siftings hopper, its location, in the average layout of stoker design, will carry it too far forward of the stoker drive toggle to insure smooth running, besides throwing the eccentric shaft at an objectionable angle, conducive to uneven and excessive wear on the drive mechanism. This is especially true where the stoker drive is of the pawl-and-ratchet type and the load on the eccentric is consequently intermittent.

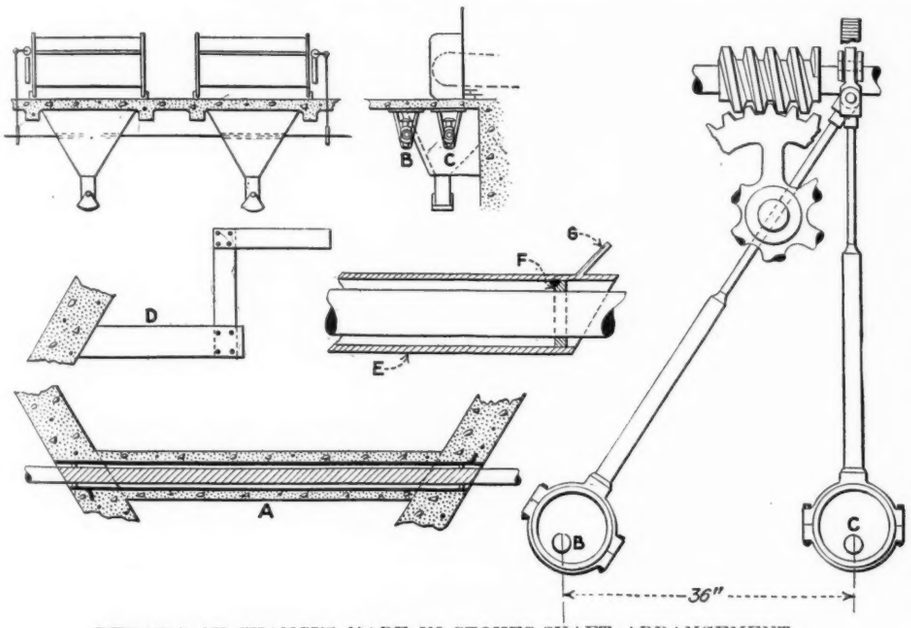
The objection to running the shaft directly through the siftings hopper are chiefly the deteriorating effect of the siftings on the steel shaft, and the waste of siftings through the lineshaft holes in the sides of the hopper. The figure *A* shows a method that I employed on a recent job when changing the shaft from the outside position *B* to the position *C*. This shaft was $2\frac{1}{16}$ in. in diameter and 160 ft. long, erected in two sections, each section serving two batteries of two 425-hp. boilers.

The device *D* was made of wood and was used to lay off the holes in the ash-hopper sides and, when held against the tunnel ceiling, gave the true elevation for all holes measuring the distance from the tunnel wall.

Eight pieces of 3-in. pipe *E* were sawed to a length equal to the width of the hopper and beveled as shown, and into each end of these was driven a babbitt collar *F*, bored to a running fit on the shaft.

The first length of shaft was then taken down and the hangers moved to the new location between the hoppers. Two of the pipes *E* were then slipped over the shaft, which was then pushed through the holes cut into the hopper sides, also slipping the eccentrics on where necessary. This was repeated until one 80-ft. section was erected. The shaft was then leveled and grouted in as shown at *A*. Wire screen was used to give the grout a bond on the pipe.

The slivers *G* were sawed into and bent back on the pipe ends to prevent the babbitt collars *F* from turning the pipes *E*. The object of the collars is to hold the shaft central of the pipes while the latter are being grouted in; they may be neglected after starting. It will be evident that in lining and leveling the shaft, the pipe, on account of the collars, will move with the shaft so that the holes in the hopper sides must be cut larger than the pipes. In this case a 4-in. diameter hole was cut for $3\frac{1}{2}$ in. outside diameter pipe. These and also the hanger-bolt holes were cut before the shaft was changed and while the boilers were in service so that the shaft could be moved in the least possible time. The new location puts the eccentric directly under the drive, and deterioration and siftings waste are taken care of.



DETAILS OF CHANGES MADE IN STOKES-SHAFT ARRANGEMENT

St. Louis, Mo. A. M. STILES.

Data of Results Wanted

A matter I would like to see taken up by *Power* readers is data relating to results obtained after suggested changes have been made. Engineers run up against some difficult problems in their work, and often outside assistance is needed in solving them. Engineering journals contain articles of value written by men who are willing to give others a lift, but the details are often too general in character.

Another thing—an engineer asks for information and other engineers answer in following issues, advising this or that, but we never know whether the information given was of benefit or what results were obtained by following out the suggestion. I, for one, would like to know the results. GEORGE J. TROSPER.

Winkleman, Ariz.

Industrial Unrest

There have been and are being written long articles in the various papers and magazines relating to industrial unrest. We hear of it in the lecture room, and it echoes from the halls of Congress. There is no single subject that is occupying the general public's mind as much, with the exception of the high cost of living, which goes hand in hand with the unrest of the masses.

The many writers and lecturers deal with the question from different angles, but it seems to me that it is all in a general way, and to my mind, we will never gain anything until we begin to deal with the matter in a more specific, or in a more individual way.

What is the real cause of this world-wide industrial unrest? Is it altogether real, or fancied? What will be the final effect if it is not soon remedied? What is the remedy and how can it be applied successfully? These are the questions that are agitating the minds of men in every rank of life.

I believe that much of the individual unrest is due entirely to a mental condition. We have heard so much of this talk, have read so much about it, we are saturated with the idea that everything is wrong in our own individual lives. A large share of our trouble is of our own making, and what we all need is to pause a few moments and take stock of ourselves, see where we are, if we have any real grievance, and get busy to try to find the remedy.

Many engineers who have talked with me recently have made numerous complaints about the conditions under which they are working, about salaries, the treatment and the lack of consideration that they are receiving. They meet in groups and discuss these matters and then separate, each in a worse frame of mind than when they met. Now an engineer has a better chance of being a deep, accurate thinker than one in any other trade or profession. Then why not, instead of meeting and grumbling about conditions, start in discussing remedies awhile?

Now, only a short time since, this feeling of unrest took hold of me. I was dissatisfied with everything in general and my position in particular. Something had to be done. After thinking over matters for several days, I decided to have a talk with the "boss." Telling other engineers my troubles did not help me any. They didn't pay my salary, and had nothing to do with the control of the place where I worked. I wanted a remedy for my troubles and there was no one that could supply it but the "boss."

I went to his office several times to see him, but he was engaged. This put me in worse humor. Finally I decided that when he wanted to see me on any matter, he sent for me to come to his office. I decided that I was as good as he was, and that this superior attitude of employers was the cause of a lot of our troubles. So the thing to do when I wanted to see him was to have him come to my office, or invite him. Shortly after reaching this decision, I met him and told him that I had been waiting to have a talk with him on some matters, and wanted him to come to my office at his first opportunity. He replied that he would. I had expected him to look surprised at the invitation, but he didn't. As I returned to the engine room, the thought struck me that it would be some time before I saw him in there, but I was mistaken again, for he was in my office within half an hour.

I told him of my dissatisfaction over several things and talked to him in his own language, using illustrations with which he was familiar. I made him see me in my position. I never had a better or more polite listener in my life. He had very little to say with the exception of asking a few questions to bring out some point more clearly. At the end of the talk, he told me that he would take the matters up with the owners and let me hear from him.

He appreciated the frank manner in which I had come to him and pointed out faults as well as bad conditions and he felt that the manner in which I had handled it was to the best interest of all concerned and that we would understand each other better in the future.

A few days after he called me to his office and more than met every condition. Now some will say that they could not talk to their superior that way. If you look on him as your superior, and he looks on you as his inferior, possibly you can't.

Now I am convinced that the most of this so-called industrial unrest is caused by absence of confidence and understanding between employer and employee. A lot of it is caused because the employee subconsciously looks on the man higher up as his superior, and imagines that the employer feels the same way. If you honestly admit that a man is better than you, it is a safe proposition that he is, or will soon be. No man is better than he thinks he is.

Several years ago, my employer at that time was a man who was said to be worth nearly a million dollars. He sent for me to come to his office to talk over some matters. The furniture in his office cost more than my entire home furnishings. He was occupying a chair that cost probably more than all the chairs I owned at that time. His clothing was of the finest material. The office was beautifully lighted. His person and surroundings simply reflected success. There I sat and talked with him, while dressed in a three dollar suit of blue overalls. When I left his presence, there was a feeling of envy in my heart, and a sneaking feeling that he was in some way my superior.

In about two months, like a clap of thunder out of a clear sky, he was declared a bankrupt. When the liabilities were printed, there was a bill for the office furnishings, one from the light company, also one from his tailor for the very clothes he had been wearing. Then the thought struck me that, although I wore overalls at the time of our conversation, they were my overalls and were paid for. Everything that I claimed I owned was mine. Although I had felt that he was above me at that time, I was his superior, morally and financially, and must be at least his equal mentally, because I was free of debt, and never again would I even secretly admit that any man was my superior, until I knew that at least the clothes he wore belonged to him and to no one else, even though he were president. And from that day I have never had any trouble talking to any employer about the matters that were important to me.

What the world needs today is, simply, increased confidence between employee and employer, and it is a matter that can only be worked out by each individual doing his part, regardless of what the other fellow says. Quit going around with a chip on your shoulder, because when you start hunting trouble, you will generally find it ready and waiting for you. And the surest way to get right is to think right.

H. A. CRANFORD.

St. Louis, Mo.

Steam Rate with Increased Back Pressure

In inquiries of General Interest for March 2, J. R. A. asks how an increase of back pressure affects the quantity of steam required by an engine. The answer was, in substance, that enough more steam should be let in to balance the higher back pressure.

It is usually thought that every pound increase in the back pressure will require an additional pound of steam per horsepower-hour, but only one-third to one-half pound of steam has to be taken from the boilers for every pound increase in the back pressure. The actual amount is determined by the degree of expansion the steam is worked to at the time it gets down to the back pressure.

Twenty years ago I was running a 15-ton can ice plant in the block belt of Alabama. The business grew and it was decided to double the capacity. The old plant made more distilled water than the compressor capacity would freeze. At this time the exhaust-steam absorbtion refrigerating machine was just coming into use. I thought that a 30-ton absorbtion machine was just what we needed, and I had figured that the economy we would get from the condensation of our exhaust steam would be velvet, and that the extra cost of operation would be that required to generate the 15 lb. of steam extra per horsepower-hour, due to the 15 lb. pressure that the generator required to make capacity.

The absorption machine was bought and installed in the off season. We got a hot week in May and our business jumped to capacity, and much to our surprise we did not have enough distilled water to fill our cans at capacity rate.

The hot wave was followed by a cool spell, and as a distilled-water ice plant is very easy to run a test on, we hunted out the trouble and found that the steam consumption did not rise pound for pound with the back pressure. Our engine driving the compressor took 29 lb. of steam per horsepower-hour, and with 15 lb. back pressure the steam rate was 36 lb. per horsepower-hour. To be sure of our results, we weighed the coal and water a number of times and got an increase of 7 to 9 lb. for the 15 lb. extra back pressure. The lower rate always was obtained when the boiler pressure was highest.

The owner was satisfied when the conditions were explained to him, and we built a cold-storage room to use up our excess capacity. JOHN H. RYAN.

Palm Beach, Fla.

Comparative Cost of Pulverized-Coal and Stoker-Fired Boiler Plants

Referring to the editorial in *Power*, March 2, entitled "Pulverized Coal at Milwaukee," and the particular statement that "the added investment in building and equipment for preparing the fuel over a stoker plant has been estimated to exceed $20 per boiler-horsepower, but the additional expenditure will be well worth while," we would like to inquire where the writer of the editorial obtained information that would lead to the conclusions given as to this difference in cost, as there is nothing like this difference. On the other hand, the costs are more nearly on a par, including building cost. We refer particularly to the actual material involved in making the complete installation; for instance, a comparison of recent estimates, based on the same horsepower of boiler equipment, shows that the cost is more favorable to the pulverized-coal installation than is the case for stoker-fired boilers, especially stokers of the underfeed type.

The plant has five 1,000-hp. boilers. The cost of the stoker plant for this type is about $150,000 and the material involved in this estimate includes the following: Underfeed stokers, complete blowing equipment for air supply, air ducts, conveying machinery outside of boiler room, conveying machinery inside of boiler room, crushers, coal bunkers, driving mechanism for stokers, blowers and conveyors, ash-handling and conveying system.

The cost of the pulverized-coal milling plant is $128,000, covering the following items to complete the installation: Separate building, coal driers, pulverizing mills, elevators and conveying machinery, blowers, dust collectors, Fuller-Kinyon pumping distribution system for pulverized coal between the boiler room and coal-mill plant (within 100 ft.), burners, feeders, coal bins for boilers, air supply for burners, motors to drive all necessary apparatus.

The only variable in this installation would be the amount of money expended on the building for the pulverized-coal plant. The figures given include a substantial structural-steel and corrugated-iron building for the coal-mill plant of sufficient size to house all the apparatus. This structure might not suit the ideas of some parties who would prefer to spend more money on the building construction, but it is beyond anybody's belief that it would be necessary or advisable to spend so much money on the building construction that it would make the comparable cost $5 per horsepower more, to say nothing of $20 per horsepower more, for a pulverized-fuel installation.

We sincerely trust you will investigate and let us have more discussion on this question as to what the writer of the editorial based his statement upon.

FREDERICK A. SCHEFFLER,
Manager Power Department,
New York City. Fuller Engineering Co.

Maximum Length of Longitudinal Joints for Boilers

Here is another suggestion on the subject of longitudinal-joint lengths for the purpose of inviting the opinion of your many readers. Quite a little printer's ink has been spread in the endeavor to determine the proper length for longitudinal seams in steam boilers, but at present it seems that we are like the circus horse that runs around a ring—we are getting nowhere.

Why not have the boiler-course lengths determined according to the plate thickness and the diameter of the shell or drum. Is it not reasonable to suppose that light plate with certain drum diameter and length of course would be likely to become affected by longitudinal cracks or grooving in less time than heavier plate?

Then why not increase the plate thickness if it is desired to increase the course length? The drum diameter should determine the course length. The greater the diameter the shorter the course, and if longer courses are desired, the plate thickness should be increased accordingly.

Engineers and boilermakers are invited to join in this "joint" debate. N. DEVERING.

Pittsburgh, Pa.

Loose Setscrew Stopped Governor

A new regulator used on a Brown engine recently failed and the safety stop preve. ed a possible wreck. The engineer served as a regulator for the remainder of the day by the use of the throttle valve until the governor was again rendered serviceable. The trouble was due to a setscrew that held one of the gears becoming loose, thus stopping the governor. This gear is now splined or keyed on, which will likely eliminate developments of a similar nature in the future.

Woburn, Mass. M. H. OSGOOD.

National Engineers' License Law

Mr. Peters, on page 434 of the March 16 issue of *Power*, suggests that when his proposed national license law goes into effect every engineer who has been successfully operating his plant for three to five years will be given a license, free of examination, to operate that particular plant.

I consider this quite logical and practical, and by doing so it allows the national engineers' license law to settle smoothly and gradually over the engineering world without any disturbance to the operation of steam plants. It also allows the engineer who is not technically qualified to prepare himself for a license.

Brenham, Tex. H. W. ROSE.

Pure Water as Feed Water

Referring to H. W. Rose's letter on page 604 of the April 13 issue on the effect of pure water on boiler metal, in which he refers to Mr. Hill's article on refrigeration on page 320 of the March 2 issue, Mr. Hill says that were it not for leakage losses the same water could be used indefinitely. Mr. Rose calls attention to the fact that distilled water has a corrosive action on boiler metal and will cause pitting. Some readers may wonder why pure water should have this effect to a much greater degree than water which has been treated, but which, of course, still contains mineral salts. S. W. Parr, of the University of Illinois, and others explain that this is due to the fact that pure water will carry a much greater content of dissolved oxygen, and that this oxygen, supplied to the tubes as the water passes through, is taken up by the metal to form iron oxide, or rust, and the unevenness of the formation will cause pitting. It is not to be understood, however, that this is a rapid process.

In this connection, it is of interest to note the description on pages 492-5 of the March 30 issue, of a 50,000-kw. power plant in Chile and the difficulties encountered in the operation. Prominent among these difficulties was the fact that the water had to be obtained by distillation of sea-water, since the rainfall is negligible and rivers are of little consequence. The plant is being successfully operated on feed water distilled from the ocean brine, and those in charge feel that part of the success is due to the purity of the water, since no scale is formed and little corrosion is experienced. From this it can be seen that the purity of the feed water is certainly not such a bugbear as Mr. Rose's letter might seem to indicate, and that the disadvantage resulting from the larger oxygen content carried by pure water might be more than overcome by the decreased amount of scale formed.

W. J. RISLEY, JR.,
Decatur, Ill. Chemical Engineer.

Care of CO_2 Recording Machines

A short time ago there was some discussion in the columns of *Power* on the merits of, and the care necessary for, recording CO_2 machines, and I would like to make a few statements from my experience, with special reference to the article by C. J. Schmid in the Feb. 17, 1920, issue.

Our machines are the latest models of a well-known make, and the men who take care of them are not only good mechanics, but also specialists on instruments, particularly on CO_2 machines, which take up the greater part of their time; but even with this expert care we cannot approach the perfection of operation which Mr. Schmid says is possible with "any mechanic."

We have ten boilers, and one man spends on an average of five hours a day on the machines for these boilers for regular routine testing, checking and adjusting, or one-half hour per boiler. Beside this, an extra half-hour or more is required for changing absorbent cartons and thoroughly cleaning the machine and lines once a week or oftener, depending on the quality of the absorbent. Moreover, we find it necessary to have a man on the night shift when several of our boilers are banked, and also on Sundays. This is so as to have the machines in order for the morning pickup of load and the several hours of heavy load which follow.

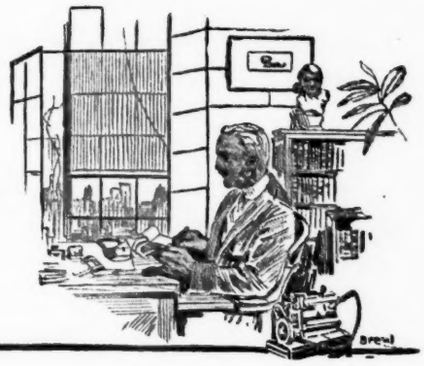
We have learned by experience that if the CO_2 machines are not looked after overnight and Sunday, one or two or more are found out of order as the boilers come on and for the time being, sometimes two or three hours, are useless, just at the time when they are needed most. And so my conclusions are, from our experience, that they do require expert care and that considerable time must be spent on them to keep them in good order. Of course any good mechanic could keep them going, but I am sure that the time required would be far in excess of the time taken by an expert, as no matter how good a mechanic he might be, he would have to take some time in following up the many lines and fittings.

We should not be misled by the word "expert" in this connection and imagine him a very high-priced man. Although he must have some ability as a mechanic, he is not necessarily a first-class mechanic, but rather a man who has specialized in the care and operation of these particular machines. He is not ordinarily as high paid as a first-class mechanic, and in our practice we do not consider it good economy to put a high-priced mechanic on the job.

Furthermore, if we did not check our machines every day against an Orsat, we probably would go along thinking that everything was well, but there is never a day when there are not a few, if not all of them, requiring adjustment. This does not mean, however, that we find CO_2 recorders impractical; they are very useful in keeping efficient fires, as a man who is experienced can judge very closely the condition of a fire by the percentage of CO_2. It would be even more useful if there were no lag in the machines, the lag being between five and ten minutes on our installation. That is where a recording pyrometer is advantageous, as it will give almost instantaneous results, the only lag being the time required for the gases to travel from the fire to the thermocouple, if that may be considered a lag. An ideal installation would be a combination of the two on each boiler, but the cost would be prohibitive.

Detroit, Mich. G. A. CHATEL.

INQUIRIES OF GENERAL INTEREST



Radial Valve Gear—What is a radial valve gear? L. C. N.

The term radial valve gear is applied to reversing mechanisms having but one eccentric or equivalent crank motion.

Jumping of Feed Pump—What causes a direct steam feed pump to make short jumps of the plunger at the beginning of the stroke? E. M.

The jumping indicates presence of air and may result from running the pump too fast, or the pump design may be such that the water is not always rising and there are pockets where air can accumulate.

Size of Pump-Suction Valves—What would be the size of suction-valve areas of a steam pump for operation at a piston speed of 100 ft. per minute? G. L. J.

The water velocity through the valve-seat open areas is generally taken as 222 ft. per min., which gives a valve area 45 per cent of the plunger area, with piston speed of 100 ft. per min. Greater liberality in valve area is conducive to good service and durability, and specifications for water-works pumps frequently demand clear valve areas 50 to 60 per cent of the plunger areas.

Pressure-Reducing and Pressure-Regulating Valves—What is the difference between a pressure-reducing and a pressure-regulating valve? R. E. T.

Any valve that effects a reduction of pressure, as by throttling the flow of a gas, vapor or liquid, would properly be called a pressure-reducing valve; and a valve that limits the discharge to a constant maximum of quantity or of pressure would be a regulating valve. A valve having the purpose of reducing steam of boiler pressure to a constant lower pressure suitable for steam heating would properly be designated as a pressure reducing and regulating valve.

Gravity, Flash and Fire Test of Lubricating Oils—What effect have gravity and flash or fire points on the lubricating value of oils? B. T.

Flash and fire points of an oil are important considerations where over-heating by bearings might result in ignition of waste or other combustible materials, and the gravity may be an index for identification of a particular kind of oil, but none of these properties can be regarded as indicative of lubricating value. The real test is the analysis which comes from observation of the effect on the friction load.

Angle of Advance and Lap for Double-Eccentric Engine—In order to obtain greater range of cutoff than 0 to ½ stroke with a double-eccentric Corliss engine, why is it necessary to set the steam eccentric with a negative angle of advance and for the steam valve to have negative lap? A. L. S.

With the Corliss valve gear the valve cannot be tripped to effect cutoff later in the stroke than when the eccentric has carried the wristplate farthest to one side of its central position. When the eccentric is set 90 deg. with the crank, the farthest swing of the wristplate to one side occurs when the piston has reached about one-half stroke, and in order for the maximum wristplate displacement to be delayed so as to permit of cutoff after one-half stroke, it is necessary for the eccentric in the beginning of the stroke to be set back of the 90-deg. position; that is, the eccentric must be set with negative angle of advance. With the eccentric thus set behind the 90-deg. position, if the valve rods were of such length as to bring the valves line on line when the wristplate was in the central position, then with the engine on a center and the wristplate hooked up to the eccentric, the edges of the valves would overlap the ports, and to obtain admission at the beginning of the stroke, it would be necessary to adjust the length of each valve rod so its valve would be brought around to a position ready to open or give the desired amount of lead at the beginning of the stroke. When the valves are thus adjusted, the negative lap, or amount of opening the valves have when the wristplate reaches its central position, is a necessary consequence of adapting the lead to negative angle of advance of the eccentric.

Water-Test Pressure on Handhole Gaskets—After washing and resetting handhole covers of a water-tube boiler, is there any objection to testing the tightness of the gasket joints with cold-water pressure pumped up to the running steam pressure for the boiler? J. R.

A pumped water pressure, unless made slowly with a hand pump, is likely to run up in excess of the intended pressure and thereby cause leaks that would not occur with the intended pressure. If gaskets have proper elasticity, and the bearing faces are true, a test-pressure of no more than half of the operating pressure should be sufficient to guarantee tight gasket joints.

Resetting Spool on Duplex Pump—On a duplex pump how would the proper position be found for resetting a crosshead spool that had become loose on one of the piston rods? H. G.

When the spool is properly set, the rocker arm which it carries will be at right angles with the piston rod when the piston rod that carried the spool is in the middle of its stroke. To place the piston rod in the middle of its stroke move the steam piston as far as it will go toward the head end of its stroke, and after making a mark on the piston rod at the end of the stuffing-box gland, move the piston as far as it will go in the other end of the cylinder and make a similar mark on the rod. After locating a mark halfway between those marks, move the piston and rod so the middle mark will be at the end of the stuffing-box gland and, with the piston thus placed in the middle of its stroke, set the crosshead to such a position that the rocker which it carries will be at right angles with the piston rod. To make the pump run the same as previously, it may be necessary to shift the position of the crosshead from the right-angle position because that position will not obtain equal lost motion on each side of the valve between the valve and valve rod operated by the rocker when the valve is in its central position.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications.—Editor.]

The Corrosion of Condenser Tubes

The fifth report to the Corrosion Committee of the Institute of Metals, England, presented at the Annual General Meeting in London, March 11, 1920, is confined to the corrosion of condenser tubes—mainly 70:30 brass. It is a study of the practical problems of corrosions in condensers under service conditions, employing either fresh or sea water, and is divided into four sections.

The problems of corrosion have been classified under five different types as outlined in the accompanying table. This classification has been made for the purpose of diagnosis of corrosion troubles, and the first section of the report deals with the methods of drawing tubes and preparing them for investigation. The appearance of tubes suffering from the various types of corrosion is discussed.

When a tube is to be withdrawn for examination, it should be carefully marked to show which is the top of the tube as it lay in the condenser and the direction of water through it. Immediately after withdrawal the ends of the tube should be plugged with tightly fitting wood plugs. The tube should be kept in a horizontal position throughout these operations and until the examination is made. When ready for examination, the tube should be cut longitudinally so as to separate the top and bottom portions and any sludge removed for chemical and microscopic examination.

After the type of corrosion has been determined, the next step is to determine the conditions that have caused the trouble.

A. Concerning the water supply:

1. Information should be obtained as to whether the water is renewed continuously or is cooled and used repeatedly. In the former case a number of chemical analyses should be made at weekly intervals. In the latter case the original supply should be analysed and *analyses made of the water actually entering the condenser* after a given length of time. This is necessary since impurities in the make-up water become concentrated due to evaporation.

2. A small sample of the water actually entering the condenser should be diverted into a settling tank and any deposit examined.

3. The material of the water mains and water ends of the condenser should be noted.

4. Information should be given as to whether the tubes are choked with solid matter and, if so, as to its nature.

5. The average water velocity through the tube should be stated.

6. The inlet and outlet temperatures at various times of the year should be given as well as information as to arrangement of baffle plates and steam distribution.

7. If failures are most pronounced at certain seasons, data should be collected as to water conditions two or three months previous to the period when the trouble commences.

B. Information concerning the tubes:

1. The composition of the tubes.

2. Length of time in service and average life of tubes in the condenser.

3. The percentage of corrosion failures of the particular batch of tubes from which samples are examined.

4. The total length of tubes.

5. Does corrosion occur at the top or bottom of the tubes, nearer the inlet or outlet, or is it irregularly distributed?

6. Is corrosion confined to a particular position in the condenser?

7. Has serious corrosion been noticed in other parts of the condenser?

8. Is corrosion present in all condensers in the plant or is it confined to one or more?

9. On land: Do failures occur at any particular time of year? and on ships: Do failures appear shortly after a period in dock or similar departure from normal routine?

10. Has any process of protection been employed?

The necessity for frequent and regular analysis of water *actually entering the condenser* is emphasized, especially when cooling towers are used. It is also pointed out the conditions which actually start the corrosive action often exist for a short time only, but that the corrosion, once started, may continue even under conditions which would not start it.

The second section is devoted to a consideration of certain features of the structure of condenser tubes which affect the problems of corrosion. Attention is directed to the presence of a surface layer of noncrystalline metal, about 0.01 mm. (0.0004 in.) which has a greater resistance to corrosion than the crystalline metal underneath. Where this layer has been penetrated corrosion proceeds at an increased rate.

The third section takes up in detail the five types or classes of corrosion.

Type I, General Thinning, is an accelerated form of the corrosion in salt solutions in which there is a gradual but uniform reduction in thickness of the tube wall. Experiments indicate that with ordinary sea water this action is so slow as to be negligible. Rapid general thinning in fresh water is usually associated with the presence of free acid. A number of tests with hydrochloric acid having a concentration of only three parts in 100,000 at ordinary temperatures showed that a reduction of 2 to 4 per cent took place in six weeks. The minute quantities of acid which will cause this trouble make it difficult to detect, especially where, as is generally the case, it enters the water supply intermittently.

TYPES OF CORROSION OF BRASS CONDENSER TUBES

Type	Name	Recognized by
I.	General thinning	Rapid thinning of tube. Little or no basic salt on tube
II.	Deposit attack	Pinhole or regional pitting beneath green salt and cuprite. Foreign deposits often accompany this type of corrosion
III.	Apparent dezincification, layer type	Layer of crystalline copper which can be readily stripped off surface of tube. Slight deposit of light green salt; sometimes a little white salt
IV.	Apparent dezincification, plug type	Copper lugs or areas beneath deposits of white salt
V.	Water-line attack	Attack similar to II, but confined to inlet end of tube

NOTE—The types of corrosion are tabulated for convenience of reference and description only. It is not intended to imply that the reactions concerned are entirely different in each case; for instance, types of attack intermediate in character between III and IV are found, and the types must be considered as merging into one another.

Type II, the so-called Deposit Attack, is usually given as the cause of pitting. In the presence of sodium chloride (NaCl) solutions the cuprous oxide (Cu_2O) on the surface of the brass gradually changes to cuprous chloride (CuCl). Where particles of this latter compound adhere to the surface of the tube they gradually change, under the influence of oxygen, to cupric chloride ($CuCl_2$) and cuprous oxide. The cupric chloride rapidly attacks the brass, oxidizing the copper and reducing the cupric chloride to cuprous chloride which in the presence of air starts the cycle of reactions again. Foreign bodies lying in the tubes help to keep the cuprous-chloride particles from being washed out with the circulating water and therefore should be kept out of the condenser as much as possible.

Type III, Layer Dezincification, is applied to that condition where layers of pure copper are found. It is stated that this is the result of disintegration of the brass tube and a redeposit of copper rather than of a removal of the zinc alone. This condition exists in both sea and fresh water and with the latter is often associated with acid water.

Type IV, Local Dezincification, is really a deposit attack, but differs from Type II. It always occurs with sea water and is associated with an adhering white salt consisting of colloidal zinc oxychloride. It is shown that the right concentration of zinc in the liquid layer next to the brass surface results in the production of this salt. Lowering the zinc content of the brass or raising that of the sea water is sufficient to prevent its occurrence.

Type V, Water Line Corrosion, in tubes partially submerged in sea water takes place, not at the water line as is commonly supposed, but sometimes as much as 2 cm.

(0.79 in.) above it. This type of attack is obviously a special form of deposit attack since it takes place under the salt deposits which are formed at, or near, the water line. It is much more severe than any of the previous types.

The fourth section of the report deals with preliminary work on electrolytic protection against deposit attack. The experiments seem to show that a current of sufficiently high density might entirely prevent corrosion. Very high densities, however, would be necessary to accomplish this.

Dr. Van H. Manning Resigns

Dr. Van H. Manning, Director of the Bureau of Mines, Department of the Interior, has tendered his resignation to President Wilson, effective June 1. Dr. Manning is leaving the Government service to accept the position as Director of Research with the recently organized American Petroleum Institute. He has been with the Department of the Interior for thirty-four years and has done much for the advancement of the mineral and coal industries and the safety of its workers. While his new duties will be in behalf of private interests, the work will, in a general way, be along similar lines to that which is being done in the Department of the Interior. In his letter to the president, Dr. Manning expressed the thought that, although the Government spends each year many millions of dollars in useful scientific work for the benefit of the people and the nation, the monetary recognition of its scientific and technical servants is not sufficient to enable them to continue their work, especially during the last few years, when it has become impossible for many men to remain in the Government service owing to the constantly increasing cost of living.

Increased Pay for Technical Men in Federal Service

The Congressional Joint Commission on Reclassification of Salaries, following a year of investigation, has submitted its report on conditions existing in the Federal Civil Service in the District of Columbia. Scientific and technical employees in the Government at Washington are among those seriously affected by the lack of uniformity in compensation. At the present time there is practically no policy governing such matters as hours of work, pay for overtime, promotions, salary advancements, etc.

The report, with its recommendations to Congress, for the first time affords the Government and employees some assurance that work involving similar duties and responsibilities will receive the same pay, that increased efficiency will be recognized by advances in salary and opportunities for promotion. Uniformity of administration is assured by making the Civil Service Commission the central employment agency for the Government and by creating a Civil Service Council composed of equal members of administrative officers and employees.

The same schedules of compensation are recommended for employees performing work requiring similar training and experience. Salaries for professional men, including those in the engineering service, the biological science service, the physical science service and the economic and political science service, range from $1,800 per annum for the junior class (men just out of college) to $5,040 per year for men in the full professional class.

During the last few years the number of resignations has grown to such proportions that it has threatened complete disruption of the service.

The Scientific-Technical Section of Federal Employees' Union No. 2, is convinced that the adoption of this report with such minor changes as may be necessary, will go far to relieve the condition now existing.

Plans for the construction of twenty-six hydro-electric power plants on the River Neckar, Germany, have practically been completed. The projects will involve an expenditure of about 35,000,000 marks, and the plants, when completed, are expected to produce a total of 50,000 kw.

Superheated Steam in Turbines

It is generally estimated that the gain in economy due to the use of superheated steam is about 1 per cent for every ten degrees of superheat. Thus, if steam having 150 deg. of superheat is employed, the resulting gain should be 15 per cent, irrespective of whether the steam is employed in a reciprocating engine or a turbine. There are certain disadvantages which follow the improper use of superheated steam, the two most important of these being the effect of high temperature on the castings of the engine, causing distortion, and the difficulty of lubrication. In the turbine the effects of distortion may be particularly serious in view of the fine clearances that have to be adopted with certain types of turbine, and it may be stated that one of the reasons why the high-speed impulse type of turbine is adopted is because with this type, owing to the high velocity of the steam, the clearances can be made more generous both in the axial and the radial direction. It is, however, advantageous to keep that portion of the turbine which is exposed to the effects of highly superheated steam as small as possible, and for this reason in the first set of nozzles the fall of pressure is often about two-thirds of the total fall between the boiler and the condenser, so that on leaving these nozzles the temperature of the steam has fallen to such a degree that its effects are greatly reduced.—*Shipbuilding and Shipping Record.*

The certificate of his recent election to honorary membership in the American Society of Mechanical Engineers was presented to Dr. Auguste C. E. Rateau on May 8 at a meeting of the Société d'Encouragement pour l'Industrie Nationale. M. Charles de Freminville, who was elected an honorary member at the same time as was Dr. Rateau, but who came to the United States to receive his certificate, pointed out at this meeting, which took the form of a discussion of the benefit that may result from frequent intercourse between the scientific and technical schools and societies of the two countries, that convenient arrangements should be made to allow American students access to French schools, while Dr. Rateau described the benefit that would result if French instructors would familiarize themselves with American ideas on the professional training of young engineers, which he thinks to be excellent, by more frequent visits and longer stays in this country.

Power development in California to meet the demands of growth will mean expenditures of approximately twenty-five million dollars per annum, according to estimates made by the Railroad Commission Engineers. Construction of four large plants which will have a capacity of 120,000 kw. is already under way. They are the Caribou Plant of the Great Western Power Co., the Kerckhoff plant of the San Joaquin Light and Power Corp., the Kern River plant of the Southern California Edison Co., and the James Plant, of Los Angeles.

Work on the new hydro-electric development at Niagara Falls, which the Hydro-Electric Power Commission of Ontario started in 1918, is progressing at an expected rate of speed. A long canal is being built to carry water around the falls and through the power plant. Some little difficulty has been met in holding the slope in the deep cuts in the earth overburden, but this has not caused serious retardation and the prospect is now that the power will be available in 1921.—*Exchange.*

Engineering research workers, on the fuel value of wood at the Forest Products Laboratory, Madison, Wis., have reaffirmed that two pounds of dry wood of any nonresinous species has the heating value of one pound of good coal. Resin gives about twice as much heat as wood, and as the average amount of resin in wood is approximately fifteen per cent, this increases the heating value thirty per cent.

New Publications

NOTES ON GAS AND OIL ENGINE ACCIDENTS. Pamphlet by the National Boiler and General Insurance Co., Ltd., Manchester, England. Paper, 5 x 7 in.; 24 pages.

This pamphlet discusses some of the more common failures of internal-combustion engine parts with regard to their causes, and gives suggestions for inspection and methods of preventing such failures.

ELEMENTS OF STEAM AND GAS POWER ENGINEERING. By Andrey A. Potter, Professor of Steam and gas Engineering, and James P. Calderwood, Professor of Mechanical Engineering, both of the Kansas State Agricultural College. Published by the McGraw-Hill Book Co., Inc., New York. Cloth; 5¼ x 8 in.; 304 pages. Price, $2.50.

This book was prepared primarily as a textbook to familiarize students with power-plant equipment before taking up the more abstract study of thermodynamics and design. The main portion of the book is divided into three parts, covering steam power, gas power and the application of steam and gas power to locomotives, trucks and automobiles. The various forms of power-generating apparatus and auxiliaries are described in a non-technical manner, and numerous illustrations are included. No attempt is made to take up the mathematical study of power engineering, although a few general principles are explained and simple problems given for study.

THE RELATION BETWEEN THE ELASTIC STRENGTHS OF STEEL IN TENSION, COMPRESSION AND SHEAR. By Fred B. Seely and William J. Putnam. University of Illinois Engineering Experiment Station Bulletin No. 115. Paper, 6 x 9 in. Price, 25 cents.

This bulletin presents the results of tests to determine the elastic shearing strength of ductile and semiductile steels and the relation between this elastic shearing strength and the elastic tensile strength, with the hope of obtaining definite information on the breakdown of elastic action and on the limits of the theories of combined stresses. From torsion tests on solid and hollow cylinders a factor is found for determining the true shearing strength from tests on solid cylindrical specimens. The ratio of this true elastic shearing strength to the elastic tensile strength is given for various grades of steel, and its bearing on the theory of combined stress is discussed. The relation between elastic tension and compression stresses and the effect of rolling on these values is considered.

Personals

L. W. Haig has been appointed district sales manager of the Packard Electric Co. at Buffalo.

George A. Shoemaker has been appointed works manager of the Bound Brook (N. J.) Oil-Less Bearing Co.,

W. L. MacFarham has been appointed general manager of the Cornwall Street Railway, Light and Power Co., Cornwall, Ontario.

Clinton E. Haver, until recently connected with Willard Case & Co., has accepted the position of works engineer with the Hooven & Allison Co., Xenia, Ohio.

John A. Lawrence, formerly assistant mechanical engineer of the New York Edison Co., has assumed the position of engineering manager for Thomas E. Murray, Inc., New York.

Herbert C. Stephens, formerly power engineer of the Northern Ohio Traction and Light Co., is now associated with the Electric Motors and Repair Co., Akron, Ohio, as sales engineer.

William A. Tietze has resigned from his position as chief engineer of the Cudahy Bros. Co., Cudahy, Wis., to become general manager of the Citizens Light, Heat and Power Co., Canby, Minn.

Lyle B. Marey, formerly superintendent of the Hart & Cooley Co., Inc., New Britain, Conn., has become connected with the Chase Companies, Inc., Waterbury, Conn., in the capacity of research engineer.

Harold D. Bliss has severed his connection as chief mechanical engineer for Morris & Co., of Chicago, Ill., and has become associated with B. P. Lientz, of the B. P. Lientz Oil Furnace Co., New York.

H. W. Jarrett, formerly assistant fuel engineer, United States Bureau of Mines, Pittsburgh, Pa., has assumed the duties of sales engineer with the George J. Hagan Co., combustion engineers, Pittsburgh, Pa.

Alfred C. Jordon, formerly superintendent of equipment for the Cumberland County Power and Light Co., Portland, Me., has resigned in order to accept a similar position with the Elmira Water, Light and Railroad Co., Elmira, N. Y.

Edward B. Richardson and Harry Gay, both members of the A. S. M. E., annuonce their partnership under the name of Richardson & Gay, Consulting Engineers. The offices of the new firm will be located at 220 Devonshire St., Boston, Mass.

Harry H. Bates has resigned as field engineer in the stoker department of the Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa., to accept the position of superintendent of equipment of the Acme White Lead and Color Works, Detroit, Mich.

Raymond E. Plimpton, formerly publication manager and field secretary of the Society of Automotive Engineers, has joined the Wales Advertising Co., of New York. His time will be devoted to the handling of advertising campaigns of a technical and semi-technical nature.

Howard I. Spohn, formerly of *Motor Age*, *Motor World*, *Automotive Industries and Commercial Vehicle*, and Frank A. Kapp, formerly of *Motor Life*, "The Automobile Trade Directory" and the "Automobile Blue Books," are now associated with the Charles H. Fuller Co., Chicago, Ill.

Charles H. Tavener, mechanical engineer of the Garden City plant of the Curtiss Aëroplane and Motor Corp., announces that he has started the practice of consulting industrial, productions and aëronautical engineering with laboratory facilities for testing and development work in New York and Boston.

Lon J. Geer has resigned as general manager of the plant of the Texas Power and Light Co. at Gainesville, Tex., and will go to Sweetwater, Tex., to become general manager of the West Texas Electric Co., a corporation which controls electric-light plants in a number of western Texas towns. H. A. Price, of Alladega, Ala., will become manager of the Gainesville plant.

Society Affairs

The New England Association of Commercial Engineers will hold its ninth annual mechanical exhibition at Worcester, Mass., July 7 to 9.

The Electric Power Club will hold its annual convention at Chattanooga, Tenn., May 5-8, with headquarters at the Signal Mountain Hotel.

The American Society of Mechanical Engineers, New Haven Branch, Connecticut Section, will hold its last meeting of the season, Friday, May 14, at the Geometric Tool Co.'s plant in Westville, Conn. Representatives of the company will present papers on "Evolution of the Opening Die Head." At 5:30 p.m. a demonstration of machines of the Geometric Tool Co. will be given at the works.

The Seventh National Foreign Trade Convention is to be held in San Francisco, Cal., May 12-15, under the auspices of the National Foreign Trade Council. A number of distinguished foreign business representatives will be present and will discuss trade opportunities in their respective countries. James H. McGraw, president of the McGraw-Hill Co., will address the convention on "The Service of the Business Press," during the group sessions on Thursday, May 13. At the fourth general session on Friday, A. C. Bedford, of the Standard Oil Company of New Jersey, will deliver an address on "Fuel Oil and Foreign Trade."

Miscellaneous News

An International Trade Fair has been organized by the Town Council of Cologne, the first event to be held on Sept. 20, with the obect of encouraging the importation of raw material into Germany.

The Public Utility Interests of New York and Pennsylvania, through a safety committee, appointed last fall, have set aside the second week of June for the observance of on accident-prevention campaign.

The Public Service Electric Co. of New Jersey, in order to meet the growing power demands in the North Jersey industrial district, is contemplating the construction of another subsidiary power plant which will double the present available supply of electric power in this section of the state. The new central station will have a capacity of 200,000 kva. and will cost approximately $25,000,000.

Business Items

The Ball Engine Co., of New York City, has moved its offices from 39 Cortlandt St., to the Vanderbilt Building, 132 Nassau St., N. Y. City.

The Strong, Carlisle & Hammond Co., Cleveland, Ohio, manufacturer of steam specialties, announces the removal of the Chicago office to 17 South Desplaines St.

The George T. Ladd Co., of Pittsburgh, Pa., manufacturers of water-tube boilers, will move their general offices and engineering department to the sixteenth floor of the First National Bank Building, Pittsburgh.

The George T. Ladd Co., Pittsburgh, Pa., manufacturer of water-tube boilers, opened a Chicago sales office at 528 McCormick Building. The new office will be in charge of W. M. McKinstry, formerly Chicago manager of the Page Boiler Co.

The Hyatt Roller Bearing Co., Industrial Bearings Division, announces the removal of its offices to 100 West Forty-first St., New York City, where larger quarters have been secured for the advertising, sales and engineering departments of the division.

The Harrison Safety Boiler Works has been incorporated under the laws of the State of Pennsylvania as the "H. S. B. W.-Cochrane Corp." The firm will specialize in the solution of problems relating to the profitable utilization of exhaust steam and the heating, metering and softening of water.

Vance McCarthy, general sales manager of the Edward R. Ladew Co., Inc., N. Y. City, manufacturers of leather belting, has been made vice president of the company. Mr. McCartny will continue in direct control of the sales organization and has retained as assistant general sales manager, Russell B. Reid.

Trade Catalogs

The Shepard Electric Crane and Hoist Co., Montour Falls, N. Y., has published a new 28-page catalog entitled "The Aerial Railway of Industry." Electric cage-operated monorail hoists of various descriptions adaptable to any industry where speedy, efficient loading and unloading of freight cars, and transportation of articles, material and apparatus from one part of the plant to another is essential are fully described and illustrated. A copy of the pamphlet will be sent on request.

The Westinghouse Electric and Manufacturing Co. has just completed publication of a new 40-page catalog entitled "Westinghouse Marine Equipment in the Fabricated Ship." The booklet describes and illustrates geared turbine machinery for 9,000-ton cargo boats built by the Merchant Shipbuilding Corporation. Photographs and line drawings indicate the type of turbines used for these ships. The detailed apparatus is exhaustively treated by means of written discussions, reproduced halftones and line drawings.

COAL PRICES

Prices of steam coals both anthracite and bituminous, f.o.b. mines, unless otherwise stated, are as follows:

ALANTIC SEABOARD

Anthracite—Coals supplying New York, Philadelphia and Boston:

	Mine
Pea	5.30
Buckwheat	$3.40@$3.75
Rice	2.75@ 3.25
Barley	2.25@ 2.50
Boiler	2.50

Bituminous—Steam sizes supplying New York, Philadelphia and Boston:

Latrobe	$4.25@$4.50
Connellsville coal	4.25@ 4.50
Cambrias and Somersets	4.35@ 5.15
Clearfields	4.00@ 4.75
Pocahontas	6.50@ 7.00
New River	6.50@ 7.00

BUFFALO

Bituminous—Prices f.o.b. Buffalo:

Pittsburgh slack	$6.00
Mine-run	6.25
Lump	6.25@ 6.50
Youghiogheny	6.75

CLEVELAND

Bituminous—Prices f.o.b. mines:

No. 6 slack	$3.25@$4.00
No. 8 slack	4.25
No. 8—¾-in.	3.50@ 4.00
No. 6 mine-run	3.50@ 4.00
No. 8 mine-run	4.50
Pocahontas—Mine-run	3.25@ 4.00

ST. LOUIS

Anthracite—Probably not more than 20 per cent of the demand in this market can be supplied. Prices effective Apr. 1 were as follows:

	Williamson and Franklin Counties	Mt. Olive and Staunton	Standard
Mine-run	$2.65@2.80	$3.00@3.50	$3.00@3.50
Screenings	2.50@2.65	2.50@3.00	2.50@3.00
Lump		3.75@4.50	3.75@4.50

Williamson-Franklin rate to St. Louis is $1.10; Other rates 95c.

CHICAGO

Bituminous—Prices f.o.b. mines:

	Freight rate Chicago
Illinois	
Southern Illinois	
Franklin, Saline and Williamson Counties	
Mine-Run	$3.00@$3.10 — $1.55
Screenings	2.60@ 2.75 — 1.55
Central Illinois	
Springfield District	
Mine-Run	$2.75@$3.00 — $1.32
Screenings	2.50@ 2.60 — 1.32
Northern Illinois	
Mine-Run	$3.50@$3.75 — $1.24
Screenings	3.00@ 3.25 — 1.24
Indiana	
Clinton and Linton Fourth Vein	
Mine-Run	$2.75@$2.90 — $1.27
Screenings	2.50@ 2.65 — 1.27
Knox County Field Fifth Vein	
Mine-Run	$2.75@$2.90 — $1.37
Screenings	2.50@ 2.60 — 1.37
Brazil Block	$4.25@$4.50 — $1.27

New Construction

PROPOSED WORK

Mass., Boston—Mowll & Rand, Archts., 50 Bromfield St., will soon award the contract for the construction of a 2 story, 150 x 260 ft. theatre and store building on Commonwealth Ave., for J. C. Kiley & Co., 18 Tremont St. A steam heating system will be installed in same. Total estimated cost, $500,000.

Mass., Boston—Mowll & Rand, Archts., 50 Bromfield St., will soon award the contract for the construction of a 12 story theatre and office building on Washington and Providence Sts., for Gorden & Shuman, 18 Tremont St. A steam heating system will be installed in same. Total estimated cost, $1,000,000.

Mass., Chicopee — Howes & Howes, Archts., 243 High St., Holyoke, will soon receive bids for the construction of 3 story housing buildings for the Dwight Mfg. Co., Front St. A steam heating system will be installed in same. Total estimated cost, $350,000.

R. I., Providence—Kendall, Taylor & Co., Archts., 93 Federal St., Boston, will soon receive bids for the construction of a 5 story, 80 x 150 ft. hospital addition for the Rhode Island Hospital, 593 Eddy St. A steam heating system will be installed in same. Total estimated cost, $300,000.

Conn., Hartford—I. A. Allen, Jr., Inc., Archt., 904 Main St., will receive bids until June 1 for the construction of a 3 story, 108 x 139 ft. school addition on Talcott St., for the city. A steam heating system will be installed in same. Total estimated cost, $600,000. W. E. Cone, 89 Asylum St., Chn.

Conn., Hartford—The City Health & Charity Depts. plan to build a heating plant and laundry for the Isolation Hospital and Almshouse on Holcomb St. Estimated cost, $80,000. Whiton & McMahon, 36 Pearl St., Archts.

Conn., Waterbury—The Connecticut Military Emergency Bd., State Armory, Hartford, will receive bids until May 15 for the construction of a 4 story, 180 x 200 ft. armory on Field St. A steam heating system will be installed in same. Total estimated cost, $400,000. Louis A. Walsh, 51 Leavenworth St., Archt.

N. Y., Brooklyn—The Bd. Educ. 500 Park Ave., New York City, received bids for the installation of heating and ventilating apparatus at P. S. 182 from Daniel J. Rice, 42nd St. and Lexington Ave., $153,330; Gillis Geoghegan, 537 West Bway., $155,300; W. J. Olvany, 177 Christopher St., $155,490. All of New York City.

N. Y., Brooklyn—The Comr. of Pub. Wks., Borough Hall, plans to build a municipal building. A steam heating system will be installed in same. Total estimated cost, $5,000,000. McKenzie, Voorhies & Gmelin, 1123 B'way., New York City, Engrs.

N. Y., Buffalo—The city received lowest bid for the installation of a refrigeration machine for the Washington Market, from the Mollenberg Betz Machine Co., 110 Washington St., at $41,830. Noted April 13.

N. Y., Coney Island (Brooklyn P. O.)—The City Comr. of Pub. Wks., Borough Hall, Brooklyn, is having plans prepared for the construction of a 4 story bath house addition. A steam heating system will be installed in same. Total estimated cost, $500,000. J. Sarsfield Kennedy, 157 Remsen St., Brooklyn, Archt. and Engr.

N. Y., Newton (Flushing P. O.)—The Bd. Educ., 500 Park Ave., New York City, will receive bids until May 12 for the construction of a high school addition on Etna Pl. A steam heating system will be installed in same. Total estmiated cost, $800,000. C. B. J. Snyder, Municipal Bldg., New York City, Archt. and Engr.

N. Y., New York—L. A. Abramson, Archt. and Engr., 43 West 46th St., is preparing plans for the consruction of a 4 story recreation center. A steam heating system will be installed in same. Total estimated cost, $150,000. Owner's name withheld.

N. Y., New York—J. Sarsfield Kennedy, Archt. and Engr., 157 Remsen St., Brooklyn, is having plans prepared for the construction of a 6 story offic building on Gold and John Sts. A steam heating system will be installed in same. Total estimated cost, $500,000. Owner's name withheld.

N. Y., Redwood—The Northern New York Utilities Inc., Light & Power Bldg., Watertown, plans to construct a sub-station here and a 3 phase high tension transmission line from here to Alexandria Bay. J. B. Taylor, Pres.

N. Y., Wende (Millgrove P. O.)—The town received bids for the installation of a heating system in the County Home, here, from John N. Danforth Co., 70 Ellicott St., Buffalo, $182,848; Power Efficiency Co., New York City, $173,200; Northwestern Heating & Plumbing Co., Erie, Pa., $177,717.

Pa., Council Bluffs—The School Bd. Independent Dist. will soon award the contract for the construction of a 2 story, 75 x 135 ft. school building. Contract for the installation of heating and plumbing systems will be sub-let. Total estimated cost, $300,000. Proudfoot, Bird & Rawson, 810 Hubbell Bldg., Des Moines, Archts.

Pa., Philadelphia—A. E. Van Bibber, 6337 Drexel Bldg., is in the market for motors, boilers and engines adaptable for the drive of paper mill machinery (used).

Md., Baltimore—R. McLaughlin, Engr., Ethington, Pa., will soon award the contract for the construction of a 2 story, 262 x 274 ft. furniture factory, power house and garage on Wilkens Ave., for the Reliable Furniture Co., 303 President St. Total estimated cost, $200,000.

Md., Catonsville—C. L. Reeder, Engr., Park Ave. and Saratoga St., Baltimore, will soon award the contract for the installation of boiler, chimney, electric generator and switchboards for the Spring Grove State Hospital, here. Total estimated cost, $100,000.

Md., College Park—C. L. Reeder, Engr., Park Ave. and Saratoga St., Baltimore, is preparing plans for the remodeling of the boiler plant for the Maryland State College, here. Estimated cost, about $50,000.

Md., Pocomoke City—The Peninsula Storage Co. is having plans prepared for the construction of a 1 story, 100 x 240 ft. cold storage plant. Estimated cost, $100,000. J. S. Nussear, 324 North Charles St., Archt.

W. Va., Charleston—The Bureau of Yards & Docks, Navy Dept., Wash., D. C., will receive bids until May 19 for furnishing and installing switchboards, oil switches and barriers, busses and connection, storage battery motor generator set, transformers, disconnecting motor starters and other electrical equipment and making connections for a complete indoor sub-station at the Naval Ordnance plant, here.

N. C., Kniston—The city plans to install an electric plant. W. C. Olsen, Sumter, S. C., Archt.

N. C., Winston-Salem—Salem College plans to build a 3 story building including a 60 x 120 ft. dining hall and a 40 x 200 ft. dormitory on Main St. A steam heating system will be installed in same. Total estimated cost, $250,000. W. C. Northup, Archt.

La., New Orleans—Loew's Enterprises, 1493 B'way., New York City, are having plans prepared for the construction of a theatre here. A steam heating system will be installed in same. Total estimated cost, $500,000. T. W. Lamb, 644 8th Ave., New York City, Archt.

Ky., Newport—St. Joseph's Orphanage plans to build a 1,000,000 gal. reservoir, and install electrical centrifugal pump and steel tank of 60,000 gal. capacity, etc. Total estimated cost, $20,000. George Hornung, 512 Fairfield Ave., Bellevue, Engr.

O., Akron—F. A. Sieberling, c/o Goodyear Tire & Rubber Co., had plans prepared for the construction of a 1 story 25 x 25 ft. Booster pumping station. Two 500,000 gal. capacity centrifugal pumps will be installed in same. Total estimated cost, $60,000. Morris Knowles, Hippodrome Bldg., Cleveland, Engr.

O., Cleveland—The city plans to construct 4 story hospital additions on Sackett Ave. and Scranton Rd. A steam heating system will be installed in same. Total estimated cost, $2,500,000. Harry L. Davis, Mayor. F. H. Betz, 604 City Hall, Archt.

O., Norwalk—Albert Kahn, Archt., Marquette Bldg., Detroit, Mich., will soon award the contract for the construction of a 2 story, 89 x 152 ft. school and parish buildings on Main and Milan Sts. for St. Paul's Parish. A steam heating boiler, etc., will be installed in separate boiler house. Total estimated cost, $200,000.

Ind., Terre Haute—The Terre Haute Atomized Fuel Co. is having plans prepared for the construction of a plant. Steam power will be installed in same. Total

estimated cost, $500,000. H. I. Gentine, 205 McKeon Bldg., Engr.

Mich., Detroit—I. M. Lewis, Archt., 503 Congress Bldg., will receive bids until May 15 for the construction of a 2 story, 100 x 200 ft. commercial building on Woodward Ave. and West Grand Blvd., for the Chinese Amer. Realty Co., 1401 Ford Bldg. A steam heating system will be installed in same. Total estimated cost, $200,000.

Mich., Detroit—The Palesting Association, 150 Fort St., West, plans to build a 4 story lodge house on Charlotte St. near Case Ave. A steam heating system will be installed in same. Total estimated cost, $500,000.

Mich., Marysville—The village plans to construct a semi-permanent water supply system consisting of pumping station with submerged intake pipe into the St. Clair River, etc. Electrically driven centrifugal pumping units will be installed in same. Total estimated cost, $200,000. Robert Gordon, Engr.

Ill., Chicago—Holabird & Roche, 104 South Michigan Ave., will soon award the contract for the construction of a 4 story, 66 x 108 ft. office building on 16th and Rockwell Sts., for J. T. Reyerson & Son, 16th and Rockwell Sts. A steam heating system will be installed in same. Total estimated cost, $350,000.

Ill., Chicago—The Illinois Malleable Iron Co., 1801 Diversy Blvd., has engaged Cahill & Douglas, Consult. Engrs., 301 Gross Bldg., Milwaukee, Wis., to prepare plans for the construction of a central power plant to replace 3 separate plants. Two 300 hp. boilers will now be installed and space arranged for 4 additional boilers. These new boilers will work in connection with four waste heat boilers, each with 4,000 sq.ft. of heating surface in connection with 4 malleable melting furnaces. The boilers in central plant will be equipped with underfed stokers and coal and ash handling equipment. The generating equipment will consist of 2 direct connected engine driven units, 1 of 350 kw. and 1 of 500 kw. capacity. An air compressor of about 1,000 to 1,500 ft. with compound steam and air cylinders will be installed in same.

Ill., Chicago—L. J. McCormick Corp., c/o Holabird & Roche, Archts., 104 South Michigan Ave., is having plans prepared for the construction of a 1 story monolithic 100 x 120 ft. theatre on Dearborn and Randolph Sts. A steam heating system will be installed in same. Total estimated cost, $1,000,000.

Wis., Milwaukee—The Sewerage Comn. will receive bids until June 3 for furnishing and erecting machinery for a sewage disposal plant in the proposed building on Jones Island. Advertised in this issue.

Wis., Sheboygan—The Helming Bros. Co., c/o Gus Helming, 1714 North 9th St., will soon award the contract for the construction of a 3 story, 60 x 150 ft. public garage on Erie Ave. Steam heating and electric power will be installed in same. Total estimated cost, $100,000.

Ia., Glidden—H. W. Porter, Secy. of the Bd. Educ., received lowest bid for the installation of a heating and plumbing system in the proposed 3 story, 90 x 162 ft. school building from the C. F. Rock Plumbing & Heating Co., St. Joseph, Mo., $36,213.

Ia., Mason City—The Miller Hotel Co. is having plans prepared for the construction of a 7-story, 132 x 165 ft. hotel. Contract for the installation of heating and plumbing systems will be sub-let. Total estimated cost, $800,000. Proudfoot, Bird & Rawson, 810 Hubbell Bldg., Des Moines, Archts.

Ia., Pocahontas—The Pocahontas Co. plans to construct a 3 story court house. A steam heating system will be installed in same. Total estimated cost, $400,000. Proudfoot, Bird & Rawson, 810 Hubbell Bldg., Des Moines, Archts.

Minn., Hibbing—John Murphy, Town Recdr., will receive bids until May 18 for the construction of a 2 and 3 story, 149 x 284 ft. recreational building on 1st Ave. and Jefferson St. A steam heating system will be installed in same. Total estimated cost, $500,000. Halstead & Sullivan, Palladio Bldg., Duluth, Archts. and Engrs.

Minn., New Ulm—The city will receive bids until May 13 for the construction of a 1 story, 40 x 90 x 90 ft. power plant, and for the installation of equipment including steam engine, exciter, etc. Total estimated cost, $200,000. A. J. Mueller, Engr. Silas Jacobson, 403 Endicott Bldg., St. Paul, Archt.

Kan., Americus—The people of Americus plan an election May 14 to vote on $18,000 bonds to construct a transmission line from here to Emporia.

Mo., Versailles—The Versailles Flour Mill & Ice Plant plans to install 130 hp. additional power in its present plant. R. Moser, Owner.

Tex., Dallas—The Security Natl. Bank, Commonwealth Bldg., had plans prepared for the construction of a 12 story, 100 x 200 ft. office building on Main and Murphy Sts. A steam heating system will be installed in same. Total estimated cost, $1,000,000. Herbert M. Greene Co., North Texas Bldg., Archt. and Engr.

Okla., Frederick—Tillman Co. is having plans prepared for the construction of a 4 story court house. A steam heating system will be installed in same. Total estimated cost, $250,000. Tonini & Bramblett, 301 Terminal Bldg., Oklahoma City, Archts.

Col., Denver—Loew's Enterprises, 1493 B'way, New York City, are having plans prepared for the construction of a theatre here. A steam heating system will be installed in same. Total estimated cost, $350,000. T. W. Lamb, 644 8th Ave., New York City, Archt.

Utah, Ogden—D. D. McKay, Engr., has filed application with the state engineer for Ogden River water rights to build a hydroelectric plant to pump water for irrigation in Weber and Boxelder Counties.

Wash., Puget Sound—The Bureau of Yards & Docks, Navy Dept., Wash., D. C., plans to furnish and install auxiliary equipment and piping for the air compressor, here. Estimated cost, $60,000.

Cal., Long Beach—Loew's Enterprises, 1493 B'way., New York City, are having plans prepared for the construction of a theatre here. A steam heating system will be installed in same. Total estimated cost, $500,000. T. W. Lamb, 644 8th Ave., New York City, Archt.

M. I., Guam—The Bureau of Yards & Docks, Wash., D. C., and the Office of Pub. Wks., San Francisco, Cal., received bids for the construction of extensions to power house, pump house, floor house and quarters at the Naval Station here, from Alfred H. Vogt, Builder's Exch., San Francisco, Cal., at $87,566; Pittsburgh Des Moines Steel Co., Munsey Bldg., Wash., D. C., at $83,700. Noted April 6.

Ont., New Toronto—The village plans to enlarge its waterworks system. Plans include the installation of a new electric pump, a new 16 in. feeder main and two 3 stage, 1,000 Imp. G.P.M. electric driven turbine pumps connected in series. Total estimated cost, $35,000. E. A. James Co., Ltd., 36 Toronto St., Toronto, Engr.

Ont., Ottawa—Stevens & Lee, Archts., 2 College St., Toronto, will receive bids for the construction of a hospital for the city. A vacuum steam heating and blower ventilating system will be installed in same. Total estimated cost, $2,500,000.

Ont., Toronto—The Bloor St. Baptist Church, Bloor and North Sts., plans to construct a church. A steam heating system will be installed in same. Total estimated cost, $500,000.

Alta., Calgary—The City Comrs. will receive bids until May 14 for the manufacture and delivery of three 200 kva., 12,000 volt to 2,300 volt transformers. J. M. Miller, City Clk.

Alta., Calgary—Loew's Enterprises, 1493 B'way., New York City, are having plans prepared for the construction of a theatre here. A steam heating system will be installed in same. Total estimated cost, $500,000. T. W. Lamb, 644 8th Ave., New York City, Archt.

Alta., Edmonton—Loew's Enterprises, 1493 B'way., New York City, are having plans prepared for the construction of a theatre here. A steam heating system will be installed in same. Total estimated cost, $400,000. T. W. Lamb, 644 8th Ave., New York City, Archt.

B. C., Victoria—Loew's Enterprises, 1493 B'way., New York City, are having plans prepared for the construction of a theatre here. A steam heating system will be installed in same. Total estimated cost, $400,000. T. W. Lamb, 644 8th Ave., New York City, Archt.

B. C., Winnipeg—Loew's Enterprises, 1493 B'way., New York City, are having plans prepared for the construction of a theatre here. A steam heating system will be installed in same. Total estimated cost, $500,000. T. W. Lamb, 644 8th Ave., New York City, Archt.

CONTRACTS AWARDED

Md., Fairfield (Baltimore P. O.)—The Globe Shipbuilding & Dry Dock Co. of Maryland, Fidelity Bldg., Baltimore, has awarded the contract for the construction of a 1 story, 30 x 110 ft. power house, 1 story, 50 x 150 ft. joiner shop, mold loft, etc., to Bancroft Jones Corp., Mutual Life Bldg., Buffalo, N. Y., at $850,000. Contract for the construction of a 1 story machine shop including boilers, etc., has been awarded to McClintic Marshall Co., Oliver Bldg., Pittsburgh, Pa., at $500,000. Noted Feb. 9.

N. C., High Point—The Harriss Corington Hosiery Mills, has awarded the contract for the construction of a 2 story, 50 x 100 ft. hosiery mill on Cox St., to J O'Connor, Washington St. A 100 hp. boiler and pipe fittings for heating system will be installed in same Total estimated cost, $35,000.

S. C., Columbia—The State has awarded the contract for the installation of a heating plant in the State Capitol, to W. B. Guimarin & Co., 1220 Washington St., at $34,480.

Kan., El Dorado—The El Dorado Hotel & Theatre Co. has awarded the contract for the construction of a 5 story, 81 x 147 ft. hotel and theatre building, to Fred Borgolte. A steam heating system will be installed in same. Total estimated cost, $300,000.

Neb., Gering—Scotts Bluff Co. has awarded the contract for the installation of a heating system in the proposed 4 story court house, to Prohs Bros., at $20,906. Noted Feb. 9.

Tex., Waco—The Central Texas Electric Ry. Co. has awarded the contract for the construction of track including grading and all trestle work on first unit of Waco Temple Interurban line from here to Robinson, to the Central Texas Eng. & Constr. Co., Amicable Life Bldg. Plans include a power house and transmission line.

Okla., Guthrie—The Scottish Rite has awarded the contract for the construction of a consistory, to James Stewart, 30 Church St., New York City. A steam heating system will be installed in same. Total estimated cost, $3,000,000.

Okla., Pawhuska—The Citizens' Natl. Bank has awarded the contract for the construction of a 7 story, 50 x 140 ft. bank, to the George M. Bliss Constr. Co., Lathrop Bldg., Kansas City, Mo. A steam heating system will be installed in same. Total estimated cost, $400,000.

Okla., Tecumseh—The State Bd. of Affairs, Oklahoma City, has awarded the contract for the construction of 5 school buildings, including a power house, to the Manhattan Constr. Co., Oklahoma City. A steam heating system will be installed in same. Total estimated cost, $152,700.

Wash., Olympia—The State Capitol has awarded the contract for the construction of a power house, to the Western Constr. Co., Seaboard Bldg., Seattle. A steam heating system will be installed in same. Total estimated cost, $100,000. Noted May 3.

Ore., Roseburg—The Riter Sanitarium Co. has awarded the contract for the construction of a 3 story, 80 x 145 ft. hospital plant, including a heating plant, cost, $20,000; administration building, cost, $10,000, to C. W. Frazier, Wilbur. Contract for installing steam heating system will be sub-let.

Cal., San Francisco—Don Lee, 1601 Van Ness Ave., has awarded the contract for the construction of a 7 story auto sales building on Van Ness Ave., to the Lindgren Co., Monadnock Bldg. A steam heating system will be installed in same. Total estimated cost, $600,000.

Ont., Acton—The town has awarded the contract for laying, jointing, trenching, backfilling and erecting valves and hydrants on cast iron pipe, to the Gartshore Thompson Pipe & Fdry. Co., Ltd., Stuart St., West, Hamilton. Electrically driven deep well centrifugal pumps will be installed in same. Total estimated cost, $30,000. Noted March 9.

Ont., Toronto—The city has awarded the contract for the installation of four 20,000,000 Imp. gal. per day centrifugal pumps for the John St. pumping station, to the Turbine Equipment Co., Ltd., C. P. R. Bldg. Estimated cost, $75,000.

Ont., Toronto—St. Michaels Hospital, Church St., has awarded the general contract for the construction of a 6 story, 100 x 180 ft. nurses residence and hospital addition on Shuler and Victoria Sts., to Archibald and Holmes, Excelsior Life Bldg. A vacuum steam heating system will be installed in same. Total estimated cost, $600,000.